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Turbine Blade Cooling Studies at Texas A&M University: 1980–2004

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Gas turbines are used extensively for aircraft propulsion, land-based power generation, and industrial applications. Developments in turbine cooling technology play a critical role in increasing the thermal efficiency and power output of advanced high-temperature gas turbine engines. Gas turbine blades are cooled internally by passing the coolant through several rib-enhanced serpentine passages to remove heat conducted from the outside surface. External cooling of turbine blades by film cooling is achieved by injecting relatively cooler air from the internal coolant passages out of the blade surface to form a protective layer between the blade surface and hot gas-path flow. The most important research contributions on turbine blade cooling studies at Texas A&M University's Turbine Heat Transfer Laboratory from 1980 to 2004 are summarized. For turbine blade internal cooling, the focus is on the effect of rotation on rotor blade coolant passage heat transfer with rib turbulators, pin fins, dimples, and impinging jets. For turbine blade external cooling, the focus is on unsteady high freestream turbulence effects on film-cooling performance with a special emphasis on turbine blade edge region heat transfer and cooling problems.

Nomenclature

A = cooling channel width
AR = cooling channel aspect ratio, $A{:}B$
B = cooling channel height
Bo = buoyancy parameter, $(\Delta\rho/\rho)Ro^2(R/D_h)$
D = cooling channel hydraulic diameter, dimple diameter
d = pin-fin diameter
e = rib height
$F_{c,\mathrm{cor}}$ = Coriolis force acting on the coolant flow
F_{cen} = centrifugal force
$F_{j,\mathrm{cor}}$ = Coriolis force acting on the jet flow
f = friction factor
f_0 = Blasius fully developed friction factor in a nonrotating, smooth tube
H = cooling channel height
h = heat transfer coefficient
L = cooling channel length; cooling channel leading surface
M = blowing ratio for film coolant, $(\rho V)_c/(\rho V)_\infty$
Nu = regionally averaged Nusselt number, hD/k
Nu_r = ribbed-side Nusselt Number
Nu_0 = Nusselt number for fully-developed, turbulent flow in a non-rotating, smooth tube
P = rib pitch
R = cooling channel rotating radius
Re = Reynolds number, $\rho VD/\mu$
Ro = rotation number, $\Omega D/V$
s = equivalent slot length for film cooling holes
T = cooling channel trailing surface
T_i = inlet bulk temperature
T_w = wall temperature
V = bulk velocity in streamwise direction
V_c = coolant velocity
V_j = jet velocity
w_b = bulk velocity in streamwise direction
x = streamwise location within cooling channel
z_0 = cooling channel streamwise location
β = angle of cooling channel orientation
$\Delta\rho/\rho$ = coolant-to-wall density ratio
δ = dimple depth
η = film cooling effectiveness, $(T-T_\infty)/(T_c-T_\infty)$
θ = nondimensional temperature, $(T-T_i)/(T_w-T_i)$ for internal cooling passage and $(T-T_\infty)/(T_c-T_\infty)$ contour temperature for shaped-hole film cooling

Introduction

GAS turbines are used for aircraft propulsion and land-based power generation or industrial applications. Thermal efficiency and power output of gas turbines increase with increasing turbine rotor inlet temperatures. Current advanced gas turbine

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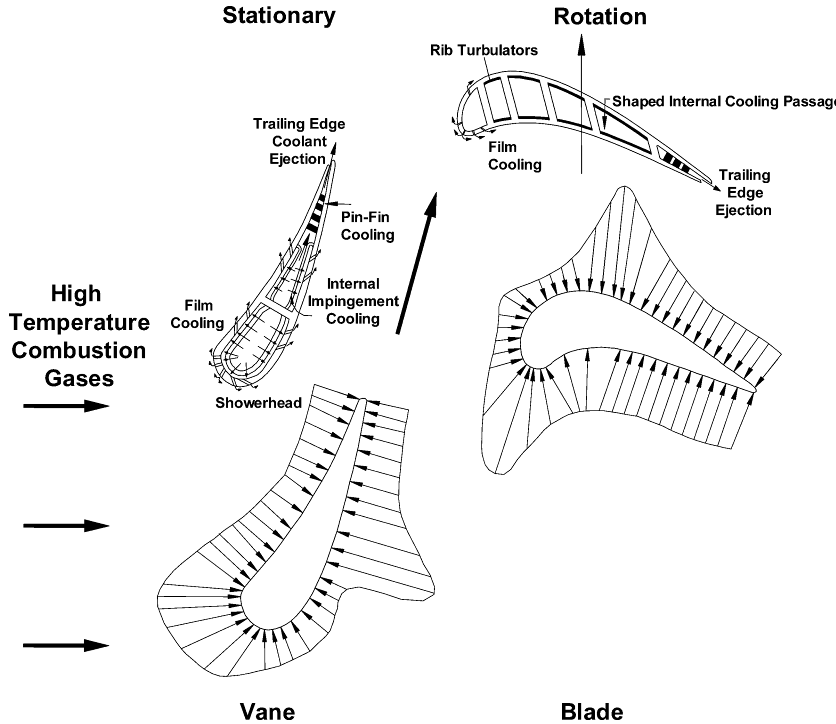


Fig. 1 Typical surface heat flux distributions and cooling schemes for a turbine blade and vane.

engines operate at turbine rotor inlet temperatures (1200–1450°C or 2200–2600°F), far higher than the melting point of the blade material; therefore, turbine blades need to be cooled. To double the engine power in aircraft gas turbines, the rotor inlet temperature should increase from today's 1450 to 2000°C (2600 to 3600°F) using the similar amount of cooling air (3–5% of compressor discharge air). For land-based power-generation gas turbines, including power generation (300–400 MW combined cycles), marine propulsion, and industrial applications such as pumping and cogeneration (less than 30 MW), the rotor inlet temperature will increase, but at a rate determined by NOx constraints; with the emphasis on NOx reduction, efficient use of cooling air becomes more important to achieve cycle efficiency gains. Therefore, high-temperature material development such as thermal barrier coating (TBC) and highly sophisticated cooling schemes such as augmented internal cooling and external film cooling are two important issues that need to be addressed to ensure high-performance, high-power gas turbines for both aircraft and land-based applications. Figure 1 shows a typical heat flux distribution on the surfaces of a turbine vane and blade and the associated internal and external cooling schemes. As the turbine inlet temperature increases, the heat transferred to the turbine blade also increases. The level and variation in the temperature within the blade material, which cause thermal stresses and blade failures, must be limited to achieve reasonable durability goals. Note that the blade life may be reduced by half if the blade metal temperature prediction is off by only 30°C (50°F). Therefore, it is critical to predict accurately the local heat transfer coefficient as well as the local blade temperature to prevent local hot spots and increase turbine blade life. Meanwhile, there is a critical need to cool the blades for safe and long-life operation.

The turbine blades are cooled with extracted air from the compressor of the gas turbine engine. Because this extraction incurs a penalty on the thermal efficiency and power output of the gas turbine engine, it is important to understand and optimize fully the cooling technology for a given turbine blade geometry under engine operating conditions. Gas turbine blades are cooled both internally and externally. Internal cooling is achieved by passing the coolant through several rib-turbulated serpentine passages inside of the blade. Both jet impingement and pin-fin cooling are also used as a method of internal cooling. External cooling is also called film cooling. Internal coolant air is ejected out through discrete holes to provide a coolant film to protect the outside surface of the blade from hot combustion gases. The engine cooling system must be designed to ensure that the maximum blade surface temperatures and temperature gradients during operation are compatible with the allowable blade thermal stress for the life of the design. Too little coolant flow results in hotter blade temperatures and reduced component life. Similarly, too much coolant flow results in reduced engine performance. The turbine cooling system must be designed to minimize the use of compressor discharge air for cooling purposes to achieve maximum benefits of the high turbine inlet gas temperature.

Research activities in turbine heat transfer and cooling began in the early 1970s; since then, many research papers, state-of-the-art review articles, and book chapters have been published. Based on more than 25 years of research experience, Han et al.[1] recently published the first book that focuses entirely on the range of gas turbine heat transfer issues and their associated cooling technologies for both aircraft and land-based gas turbines. In addition, several recent publications that review state-of-the-art papers on worldwide turbine blade cooling and heat transfer research activities are available. These include internal convection heat transfer and cooling by Han and Dutta,[2] turbine cooling and heat transfer by Lakshminarayana,[3] rotor coolant passage heat transfer by Dutta and Han,[4] recent developments in turbine blade film cooling by Han and Ekkad,[5] recent developments in turbine blade internal cooling by Han and Dutta,[6] a symposium volume dealing with heat transfer in gas turbine systems by Goldstein,[7] a detailed review of convective heat transfer and aerodynamics in axial flow turbines by Dunn,[8] and recent studies in turbine blade cooling by Han.[9] Interested readers can gain information from these publications. However, the 2004 AIAA Thermophysics Award Lecture is limited only to a review of the most important research contributions on turbine blade cooling studies at Texas A&M University from 1980 to 2004. The present paper does not intend to review many other researchers' papers in this field.

Turbine Blade Internal Cooling

Gas turbine cooling technology is complex and varies between engine manufactures. Figure 2 shows the typical cooling technology with three major internal cooling zones in a turbine rotor blade with strategic film cooling in the leading edge, trailing edge, pressure and suction surfaces, and blade tip region. The leading edge is cooled by

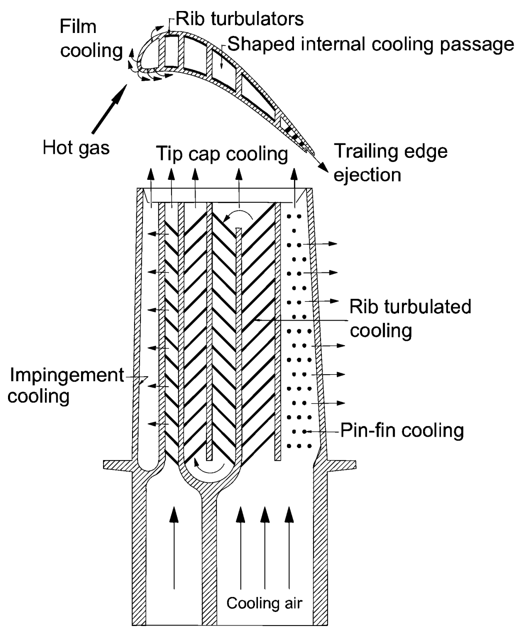


Fig. 2 Typical techniques for turbine blade internal cooling.

jet impingement with film cooling, the middle portion is cooled by serpentine rib-turbulated passages with local film cooling, and the trailing edge is cooled by pin fins with trailing-edge injection. Jet impingement cooling can be used mostly in the leading-edge region of the rotor blade, due to structural constraints on the rotor blade under high-speed rotation and high centrifugal loads. Practically, a blade midchord region uses serpentine coolant passages with rib turbulators on the inner walls of the rotor blades, whereas the blade trailing-edge region uses short pins due to space limitation and structural integration. Heat transfer in rotating coolant passages is very different from that in stationary coolant passages. Both Coriolis and rotating buoyancy forces can alter the flow and temperature profiles in the rotor coolant passages and affect their surface heat transfer coefficient distributions. It is very important to determine the local heat transfer coefficient distributions in the rotor coolant passages with impingement cooling, rib-turbulated cooling, or pin-fins cooling under typical engine cooling flow, coolant-to-blade temperature difference (buoyancy effect), and rotating conditions. It is also important to determine the associated coolant passage pressure losses for a given internal cooling design. This can help in designing an efficient cooling system and prevent local overheating on the rotor blade.

Coolant Passage Heat Transfer with Rib Turbulators

In the early 1970s, cooling designs of advanced gas turbine blades, repeated transverse ribs (orthogonal ribs or 90 deg ribs, oriented 90 deg to the coolant flow) are cast on two opposite walls of internal cooling passages to enhance heat transfer. Thermal energy conducts from the external pressure and suction surfaces of turbine blades to the inner zones, and that heat is removed by internal cooling. The internal cooling passages are mostly modeled as short, square or rectangular channels with different aspect ratios. The heat transfer performance in a nonrotating cooling channel with two opposite repeated rib-roughened walls primarily depends on the rib configuration, the channel aspect ratio, and the flow Reynolds number. There have been several fundamental studies (e.g., Webb et al.[10] and Han et al.[11]) to understand the heat transfer enhancement phenomena by the flow separation–reattachment tripped by repeated transverse ribs (turbulence promoters, rib turbulators, trip strips). These flow separations reattach the boundary layer to the heat transfer surface, thus increasing the heat transfer coefficient. Moreover, the separated boundary layer enhances turbulent mixing, and therefore, the heat from the near-surface fluid can more effectively be dissipated to the main flow, thus increasing the heat transfer coefficient. Repeated transverse ribs mostly disturb only the near-wall flow for heat transfer enhancement, consequently, the pressure drop penalty by repeated transverse ribs is affordable for turbine blade internal cooling designs. In general, repeated transverse ribs used for early turbine cooling designs are nearly square in cross section with a typical relative rib height of 5–10% of channel hydraulic diameter and with a rib spacing-to-height ratio varying from 5 to 15 (Han et al.[11]).

Angled Ribs

Han et al.[11] studied heat transfer and friction for rib-roughened surfaces. This was the first paper in open literature proposing that the repeated angled ribs (inclined ribs or skewed ribs, oriented 45 deg to the coolant flow) performed better than the repeated transverse ribs in a parallel-plate channel. They found that heat transfer augmentation in a parallel-plate channel with repeated 45-deg angled ribs was about same as with repeated 90-deg transverse ribs, but pressure drop was dramatically reduced. This important finding could be applied for general heat transfer enhancement devices. In the 1980s, Han,[12] Han et al.,[13] Han,[14] Han and Park,[15] Han et al.,[16] and Park et al.[17] systematically investigated heat transfer enhancement in rectangular channels with turbulence promoters. They were the first group in the open literature proposing that the repeated angled ribs (oriented 30, 45, or 60 deg to the coolant flow) performed better than the earlier repeated transverse ribs over a wide range of rectangular channels (where channel aspect ratio varied 4/1, 2/1, 1/1, 1/2, and 1/4) applicable for gas turbine blade cooling designs. They found that the ribbed side heat transfer enhancements are about three times and the pressure drop penalties are about four to eight times the values for 45- and 60-deg ribs compared to a smooth channel for Reynolds numbers between 1.5×10^4 and 6×10^4. The pressure drop penalties are only two to four times for the 45- and 60-deg angled ribs with the same level of heat transfer enhancement for the narrow aspect ratio channels (aspect ratios 1/2 and 1/4 applicable to cooling channels near leading-edge region of the blade). However, for the same level of heat transfer enhancement in a broad aspect ratio channel (aspect ratio 4/1 applicable to cooling channels near the trailing-edge region of the blade), the pressure drop penalties are as high as 8–16 times the friction factor in a smooth channel for angled ribs. They concluded that the narrow aspect ratio channel performs better than a broad aspect ratio channel with angled ribs. They developed heat transfer and friction correlations for a wide range of rib turbulators geometry (such as size, angle, shape, and distribution), rectangular cooling channel aspect ratio, and flow Reynolds number. These correlations have been widely cited by turbine cooling researchers as baseline data comparison and used by turbine cooling designers for advanced heat transfer analysis and optimal cooling design prediction for newly developed turbine blades. It is fair to say that all major turbine manufacturers, such as General Electric, Pratt and Whitney, Siemens–Westinghouse, ABB, and Solar Turbines, utilize this new angled-rib cooling concept to replace earlier transverse-rib configurations for improving their turbine blade internal cooling designs.

Effect of Rib Angle and 180-Degree Sharp Turn

Serpentine channels are common in typical turbine blade cooling designs. In the 1980s, Chandra et al.[18] and Han and Zhang[19,20] studied the effect of rib angle orientation on local heat transfer distribution in a three-pass rib-roughened channel. They found that the rib angle and orientation and the sharp 180-deg flow turn significantly affected the local heat transfer and pressure drop distributions. The combined effects of these parameters increased or decreased the heat transfer coefficients after the sharp 180-deg flow turns. The angled ribs, in general, provided higher heat transfer coefficients than the transverse ribs, and the parallel angled ribs gave higher heat transfer than the crossed angled ribs. However, the 180-deg flow turn direction and angled rib orientation can cause a reduction in heat transfer. They proposed that care needs to be taken in angled rib alignment in the flow turn regions, and guidance in that respect is provided in the discussed results for turbine blade cooling designs.

High-Performance V-Shaped and Delta-Shaped Ribs

In 1990s, Han et al.,[21] Lau et al.,[22,23] Han and Zhang,[24] and Han et al.[25] further proposed the high-performance three-dimensional turbulence promoters, such as V-shaped ribs, broken parallel and V-shaped ribs, and delta-shaped vortex generators for advanced turbine cooling system designs, as shown in Fig. 3. They presented the Nusselt number ratio of ribbed side vs the friction factor ratio (heat transfer enhancement vs pressure drop penalty, i.e., thermal performance curve) for Reynolds numbers between 1.5×10^4 and 8×10^4. They found that the ribbed side Nusselt number ratios, that is, heat transfer augmentation, for 45- and 60-deg V-shaped ribs are higher than the earlier mentioned 45- and 60-deg angled ribs. This is because the V-shaped ribs induce four-cell vortices as compared to the two-cell vortices induced by the angled ribs as shown in Fig. 4. In addition the ribbed side Nusselt number ratios, that is, heat transfer enhancement, for 45- and 60-deg broken angled ribs or broken V-shaped ribs are much higher than the corresponding 45- and 60-deg nonbroken angled ribs or nonbroken V-shaped ribs. However, the corresponding friction factor ratios are comparable with each other for broken and nonbroken rib configurations, as shown in Fig. 5. They also reported on high-performance delta-shaped ribs, forward or backward aligned. The isolated three-dimensional projections disturb the boundary layer and create vortices along the delta-shaped faces. Highest Nusselt number ratios (heat transfer enhancement) are obtained with backward aligned delta-shaped ribs. The backward delta-shaped ribs, the overall best performer in the group, produced three to four times heat transfer augmentation over

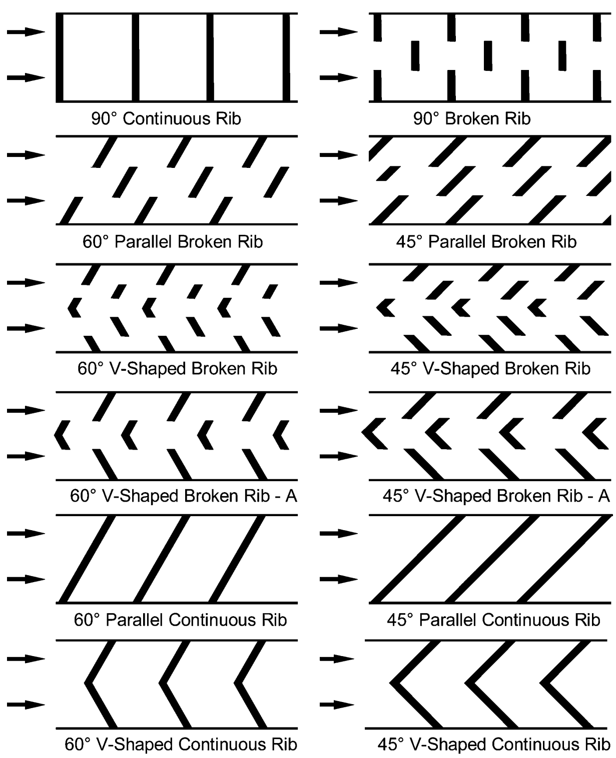


Fig. 3 High-performance rib turbulators for turbine blade internal cooling from Han and Zhang.[24]

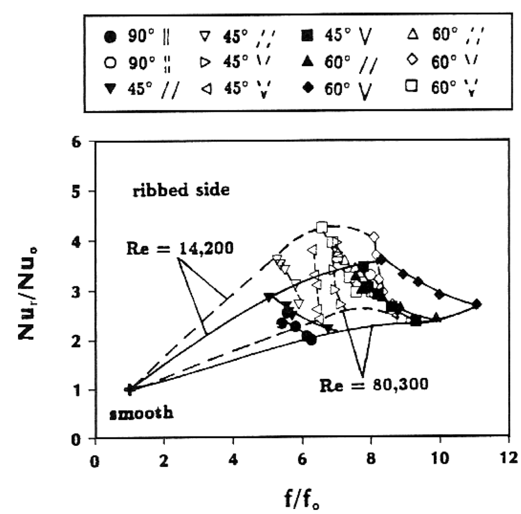


Fig. 5 Comparison of heat transfer performance for broken and nonbroken rib configurations from Han and Zhang.[24]

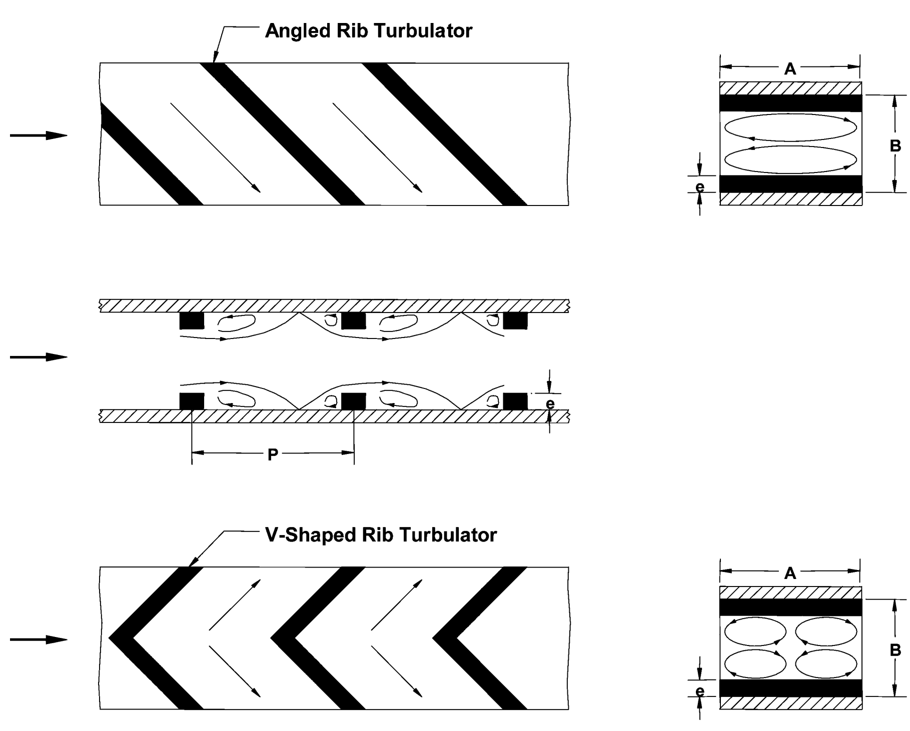


Fig. 4 Angled and V-shaped rib-induced secondary flow.

a smooth surface result and the pressure drop was seven to nine times higher. These promising V-shaped or delta-shaped ribs have being considered to be integrated into the new generation turbine blade cooling designs.

Development of Measurement Techniques

In the 1980s, Lau et al.,[26] Han et al.,[27] and Lau et al.[28] applied the existing naphthalene sublimation mass transfer measurement technique for the detailed heat transfer coefficient distributions in augmented cooling channels with rib turbulators and pin fins. One of significant findings by using the detailed naphthalene sublimation technique was the earlier mentioned combined effect of rib angle and sharp 180-deg flow turn on the local heat transfer coefficient distributions in a two-pass ribbed channel. This is one of the very powerful tools in determining detailed heat transfer coefficient distributions in cooling channels with repeated ribs or pin fins. In 1990s, Ekkad and Han[29] Huang et al.,[30] Ekkad et al.,[31,32] and Ekkad and Han[33] developed an innovative measurement technique, the transient liquid crystal imaging method, to determine very detailed heat transfer coefficient mappings in augmented cooling channels with rib turbulators (Fig. 6) and impinging jets (Fig. 7), as well as to determine very detailed surface heat transfer coefficient and film-cooling distributions on the film-cooled blades. The transient liquid crystal imaging method has become a very powerful tool for researchers in quantitatively determining local heat transfer distributions and by designers for identifying local hot spots in complex internal cooling passages, as well as on external film cooling surfaces, which are critical in determining the turbine blade durability and life. Many significant findings by using the transient liquid crystal image technique for both internal cooling channels as well as external blade film cooling will be addressed in this paper.

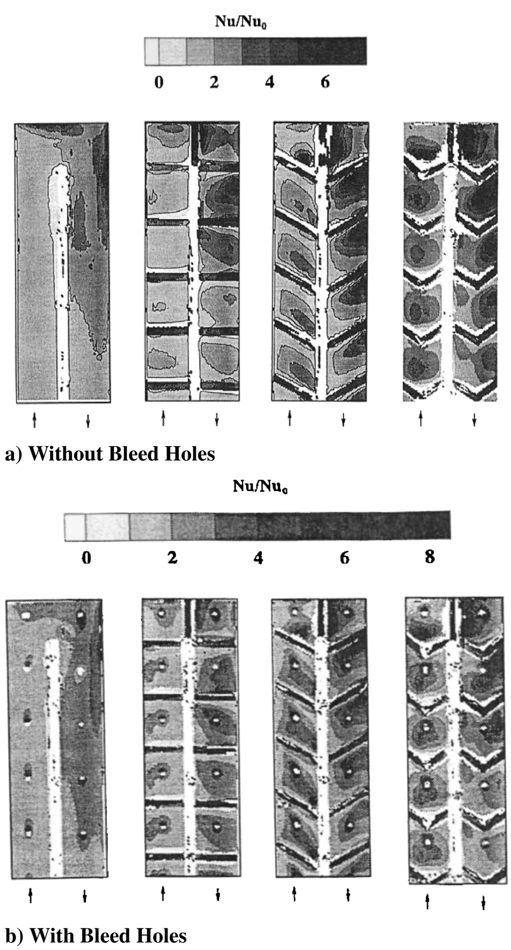


Fig. 6 Heat transfer enhancement in cooling passages with and without film-coolant extraction holes from Ekkad and Han[29] and Ekkad et al.,[34] left to right: smooth, 90-deg parallel, 60-deg parallel, and 60-deg V-shaped turbulators.

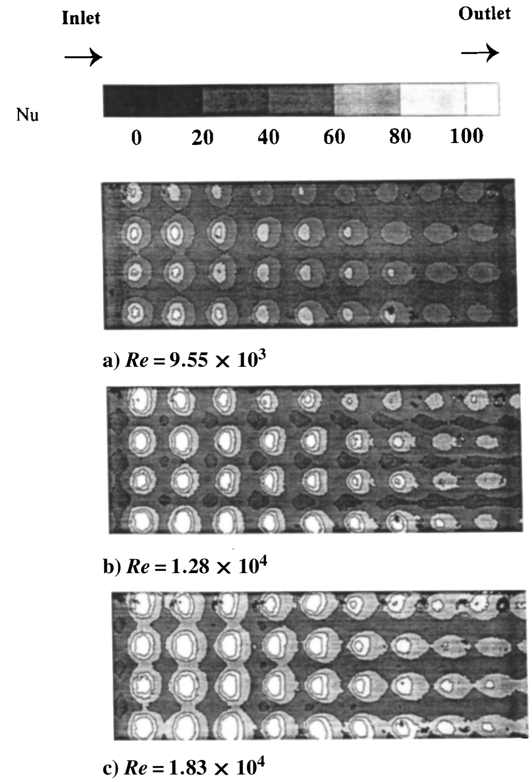


Fig. 7 Spent air crossflow effect on impingement cooling heat transfer coefficient distributions from Huang et al.,[30] left, $X/d = 3.5$ and right, $X/d = 44.5$.

Effect of Film Cooling Holes

Most modern turbine blades have repeated ribs in the internal coolant channel and film cooling for the blade surface. Therefore, some of the cooling air is bled through the film-cooling holes for blade surface protection from high-temperature combustion gases. The presence of periodic ribs and bleed holes creates strong axial and spanwise variations in the heat transfer distributions on the internal cooling channel surface. In the 1990s, Ekkad et al.[34,35] studied the detailed heat transfer coefficient distributions with different rib orientations in a two-pass channel with and without repeated film-cooling bleed holes. Figure 6 shows the detailed heat transfer distributions (Nusselt number ratios and heat transfer enhancement) in two-pass channels with ribs and bleed holes by using the earlier mentioned transient liquid crystal image technique. They show that the Nusselt number ratios for the 60-deg angled ribs are higher than those for the 90-deg transverse ribs and subsequently higher than the smooth channel results. The heat transfer coefficient distributions with the ribs show that the bleed holes increase heat transfer coefficient in the near-hole regions, but no broader impact by these bleed holes is noticeable in these results. They found that the regional-averaged Nusselt number ratios (heat transfer enhancement) for different rib orientations are almost identical with and without bleed-hole extraction. This indicates that 20–25% reduction of the coolant flow can be used for blade film cooling without significantly affecting the ribbed channel internal cooling performance.

Compound Internal Cooling Techniques

Several internal heat transfer enhancement techniques have been discussed in preceding sections. The most common methods of heat transfer augmentation in gas turbine blades are ribs, pins, and impinging jets. It is shown that these enhancement techniques increase heat transfer coefficients, but can combining these techniques increase the heat transfer coefficient more? Zhang et al.[36–38] studied the heat transfer and friction in rectangular channels with a rib and groove combination. The Stanton numbers for the ribbed–grooved walls are higher than those for the only-ribbed walls at similar rib spacing values. They also used different types of inserts to study the combined rib (tripping boundary layer) and twisted tape inserts (creating swirl flow) in circular tubes and square ducts. Four test configurations were used: twisted tape, twisted tape with interrupted ribs, hemicircular wavy tape, and hemitriangular wavy tape. Twisted tape with interrupted ribs, comparable with the hemitriangular wavy tape, provided a higher overall heat transfer performance over twisted tape without ribs and hemi circular wavy tape. Akella and Han[39] and Azad et al.[40,41] studied the combined effect of impinging jets on cooling channels with angled ribs, pin fins, or dimples. They found that the angled ribs are more effective in enhancing jet impingement heat transfer at higher jet Reynolds number flows. It can be argued that the crossflow developed by spent jets is stronger for higher jet Reynolds numbers and the angled ribs are more effective in heat transfer augmentation with a stronger crossflow. They also found that the pinned surface performs better than the dimpled surface at a lower jet Reynolds number. However, at higher jet Reynolds numbers, the dimpled surface performs better than the pinned surface for a certain spent flow orientation.

Summary

Today, rib-turbulated cooling is a major cooling technology for turbine blades as well as a very active research area for convective heat transfer augmentation in general. Han and his coworkers' pioneering work on this subject in the 1980s is now a standard reference and has paved the way for many researchers who are following their work. They have published more than 45 journal papers (with 37 papers cited in this section) in the area of heat transfer augmentation with various rib turbulators, vortex generators, pin fins, dimples, and impinging jets. This large number of journal papers concentrated in one field has documented excellent references for the enhanced heat transfer research community. Not only have turbine cooling designers and researchers heavily used their work, but also their papers have often been referred by worldwide researchers and engineers who need to augment their heat transfer devices such as those for aerospace, solar collector, nuclear fuel element, fusion blanket cooling, electronic cooling, and general heat exchanger applications.

Effect of Rotation on Turbine Blade Internal Cooling

In modern gas turbine blades, cooling air is circulated through internal serpentine cooling passages to remove heat from the blade. Coolant flow analysis in rotor blades has another dimension added by rotational forces, and these rotational effects in a coolant passage of a rotor blade significantly affect the heat transfer distribution. Rotational effects on the coolant passage heat transfer were not fully recognized until the late 1980s and early 1990s. Both Coriolis and rotating buoyancy forces can alter the flow and temperature profiles in the serpentine coolant passages and affect their surface heat transfer coefficient distributions. Rotation-induced secondary flows in a rotating two-pass channel are different for the first pass radial outflow and second pass radial inflow. Because the direction of the Coriolis force is dependent on the direction of rotation and flow, the Coriolis force has a different direction in the two passages. The rotation direction remains the same for the two channels, but the direction of flow gets reversed from the first channel to the second in the 180-deg turn. Therefore, the direction of the Coriolis force is opposite in these two channels, and the resultant rotation-induced secondary flow is different. Figure 8 shows the combined effects of Coriolis and rotational buoyancy on flow distribution by Han et al.[42] For radial outward flow in the first channel, the Coriolis force shifts the core flow toward the trailing wall. If both trailing and leading walls are symmetrically heated, then faster moving coolant near the trailing wall would be cooler than the slow moving coolant near the leading wall. Rotational buoyancy is caused by a strong centrifugal force that pushes cooler heavier fluid away from the center of rotation. In the first channel, rotational buoyancy affects the flow in a fashion similar to the Coriolis force and causes a further increase in flow near the trailing wall of the first channel, whereas the Coriolis force favors the leading side of the second channel. The rotational buoyancy in the second channel has the effect of making the flow distribution more uniform in the duct. Figure 9 shows the predicted velocity and temperature distribution in a square channel with radial outward flow. These predictions of Dutta et al.[43] include both the Coriolis and rotational buoyancy effects in the momentum equation and a modified k–ε turbulence model.

Rotational Effect on Coolant Passage Heat Transfer

Rotation induces Coriolis and centrifugal forces that produce cross-stream secondary flow in the rotating coolant passages; therefore, heat transfer coefficients in rotor coolant passages are very

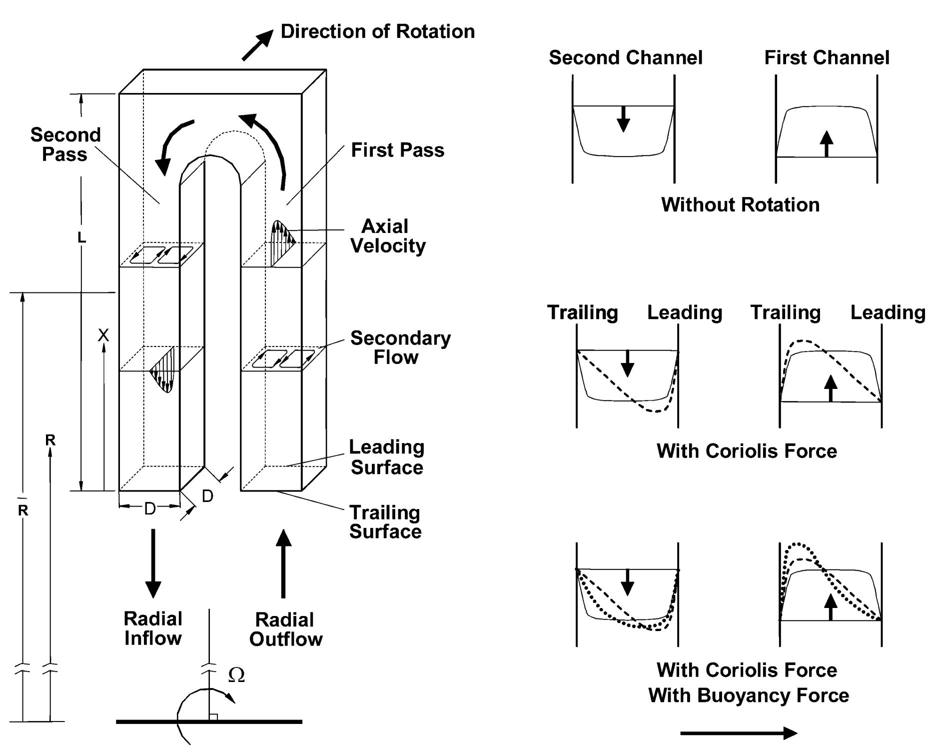


Fig. 8 Coolant flow through a two-pass rotating channel from Han et al.[42]

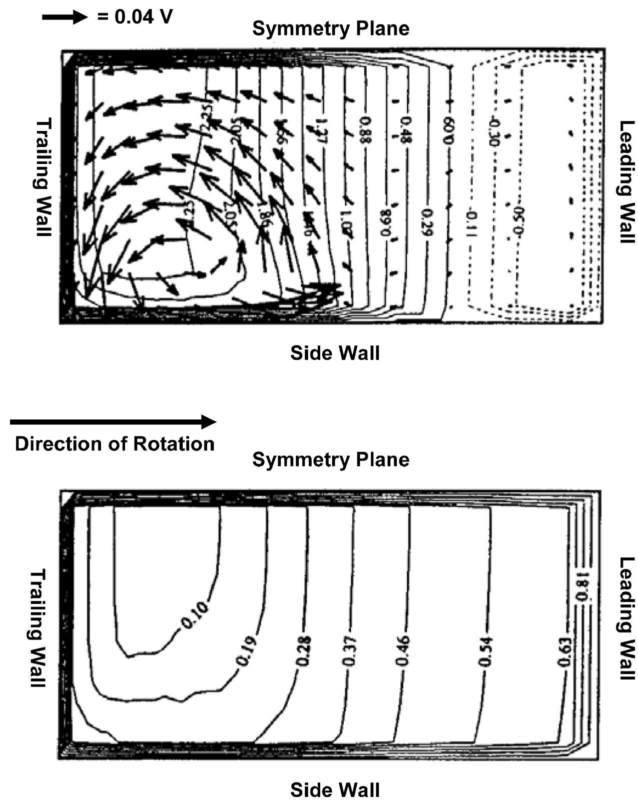


Fig. 9 Predicted velocity and temperature distributions in a square channel with radial outward flow from Dutta et al.[43]

much different from those in nonrotating channels. There are only limited papers available in open literature that study the rotational effect on rotor coolant passage heat transfer, for example, several pioneer papers published by researchers from the United Technologies Research Center (Wagner et al.[44,45] and Johnson et al.[46,47]) reviewed and cited in Chapter 4 of Han et al.[1] Wagner et al.[44,45] and Johnson et al.[46,47] studied heat transfer in rotating multipass coolant passages with square cross section and smooth walls at uniform wall temperature conditions. Results show that the heat transfer can enhance two to three times on the trailing surface and reduce up to 50% on the leading surface for the first-pass radial outward flow passage; however, the reverse is true for the second-pass radial inward flow passage due to the flow direction change. Without consideration of the rotational effect, the coolant passage would be overcooled on one side while overheated on the opposite side. Results also show that the heat transfer difference between leading and trailing surfaces is greater in the first pass than that in the second pass due to the centrifugal buoyancy opposite to the flow direction. They also studied heat transfer in rotating multipass coolant passages with square cross section with 45-deg rib-turbulated walls at uniform wall temperature conditions. Results show that rotation and buoyancy, in general, have less effect on the rib-turbulated coolant passage than on the smooth-wall coolant passage. This is because the heat transfer enhancement in the ribbed passages is already up to 3.5 times higher than in the smooth passages; therefore, the rotational effect is still important but with a reduced percentage. Results also show that, like a nonrotating channel, the 45-deg skewed ribs perform better than 90-deg transverse ribs and subsequently better than the smooth channel.

Nonuniform Wall Temperature Effect

From the preceding analyses, the rotation effect on channel heat transfer comes from the Coriolis and centrifugal forces. The centrifugal force is known as rotation buoyancy when there is a temperature difference between the coolant and the channel walls at rotating conditions. Because the temperature difference between the coolant and the channel walls varies along the coolant passages in real turbine cooling applications, so does the rotation buoyancy. Therefore, it is expected that the channel wall heating conditions would affect rotor coolant passage heat transfer. Han and Zhang[48] and Han et al.[49] studied the uneven (nonuniform) wall temperature effect on rotating two-pass square channels with smooth walls. They concluded that, in the first pass, the local uneven (non uniform) wall temperature interacts with the Coriolis force-driven secondary flow and enhances the heat transfer coefficients in both leading and trailing surfaces, with a noticeable increase in the leading side, as compared with the uniform wall temperature case. However, the uneven wall temperature significantly enhances heat transfer coefficients on both leading and trailing surfaces. Parsons et al.[50] and Zhang et al.[51] studied the influence of wall heating condition on the local heat transfer coefficient in rotating two-pass square channels with 90- and 60-deg ribs on the leading and trailing walls, respectively. They concluded that the uneven (nonuniform) wall temperature significantly enhance heat transfer coefficients on the first-pass leading and second-pass trailing surfaces as compared with the uniform wall temperature condition.

Channel Orientation Effect

Because the turbine blade is curved, the rotor blade cooling passage can have different channel orientations with respect to the rotating plane. Researchers from the United Technologies Research Center[47] studied the effects of rotation on the heat transfer for smooth and 45-deg ribbed multipass square channels with channel orientations of 0 and 45 deg to the axis of rotation. They found that the effects of Coriolis and buoyancy forces on heat transfer in the rotating passage are decreased with the passage at 45 deg compared to the results at 0 deg. This implies that the difference in heat transfer coefficients between the leading and trailing surfaces due to rotation is reduced when the passage has an angle to the axis of rotation. Parsons et al.[52] used 60-deg angled ribs, and Fig. 10 shows the channel orientation with respect to the rotation axis influences the secondary flow vortices induced by rotation. Dutta and Han[53] used high-performance broken V-shaped ribs, and Fig. 11 shows the secondary flow developed by broken V-shaped ribs could interact with the secondary flow of rotation and a new flow pattern might be established. They concluded that a turbine-cooling channel orientation

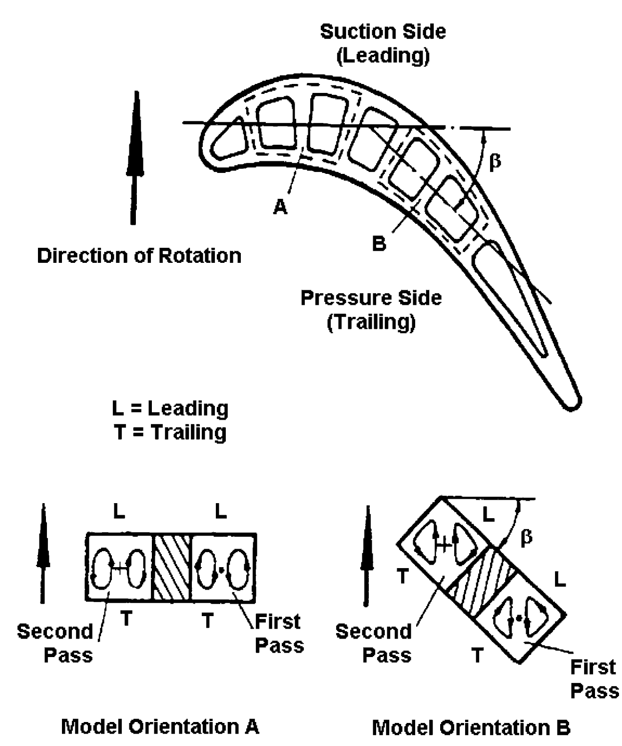


Fig. 10 Schematic of cooling channel orientations in an airfoil used by Parsons et al.[52]

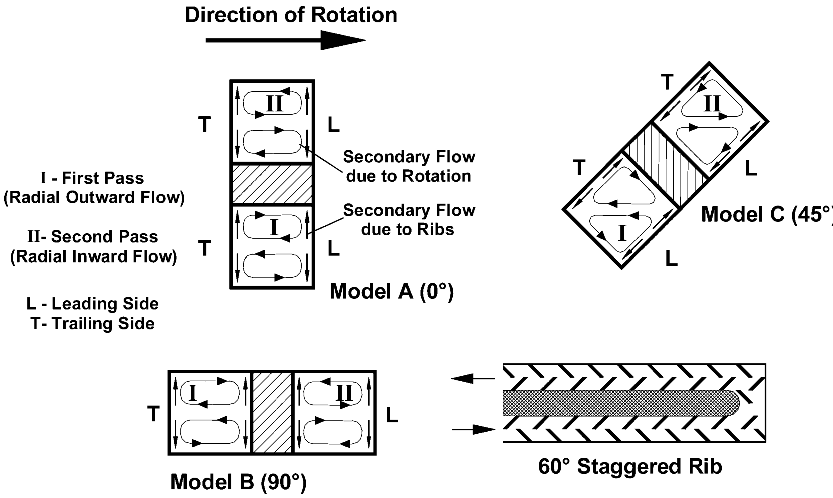


Fig. 11 Secondary flow vortices induced by rotation, channel orientation, and broken V-shaped ribs used by Dutta and Han.[53]

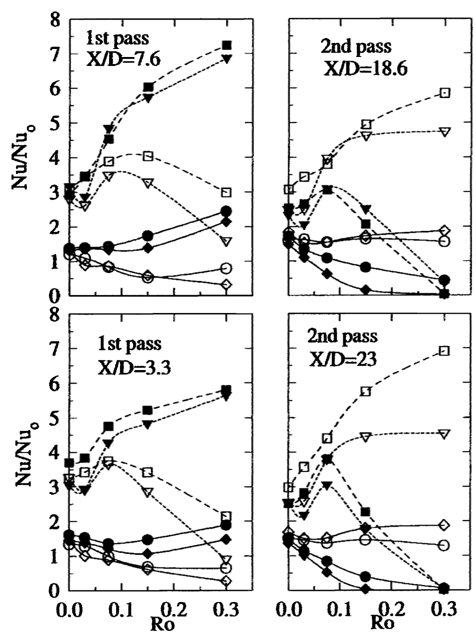


Fig. 12 Effect of rotation and channel orientation (models A and B) on heat transfer enhancement in smooth and ribbed channels used by Dutta and Han[53]: surface roughness, model A, leading, ○, smooth duct and ▽, ribbed duct, trailing, ●, smooth duct and ▼, ribbed duct; model B, leading, ◇, smooth duct and □, ribbed duct and trailing, ◆, smooth duct and ■, ribbed duct.

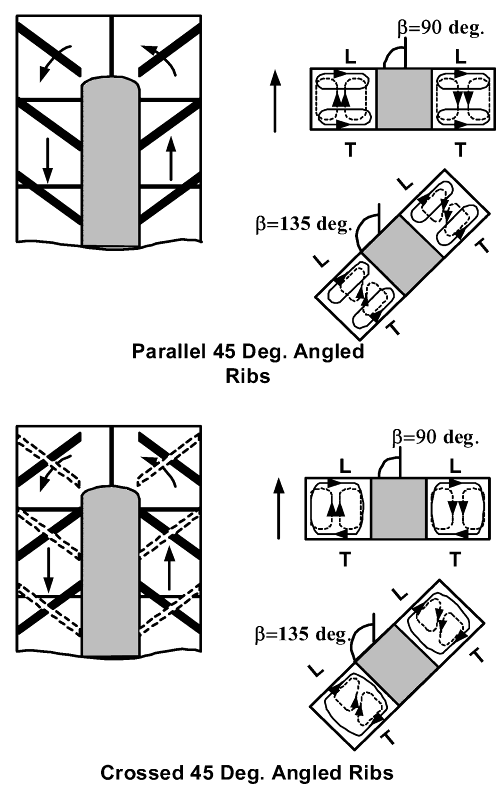


Fig. 13 Secondary flow vortices induced by rotation and channel orientation in two-pass square duct from Al-Hadhrami and Han.[54]

can be redesigned so that heat transfer on both the pressure and suction sides of the blade-cooling channel can be enhanced due to rotation, as shown in Fig. 12. Al-Hadhrami and Han[54] used parallel and crossed 45-deg angled ribs in rotating two-pass square channels to study the effect of channel orientation on heat transfer. Figure 13 shows the two cell vortices induced by rotation, the two cell vortices induced by the parallel 45-deg ribs, and the single-cell vortex induced by the crossed 45-deg ribs. They concluded that the broken V-shaped ribs are better than the 60-deg angled ribs and the parallel 45-deg angled ribs are better than the crossed 45-deg angled ribs. In general, the difference between leading- and trailing-wall heat transfer coefficients is reduced for the channel with a 45-deg angle to the axis of rotation. Rotating coolant passages with triangular cross section might be used on some portion of the blade to provide compact channel structure and good cooling efficiency. Dutta et al.[55,56] studied the effect of rotation on the heat transfer coefficients in two-pass triangular ducts with smooth and ribbed walls. They also found that channel orientation and uneven wall temperature have a significant effect on the surface heat transfer coefficient distributions.

Channel Aspect Ratio and Orientation Effect

Most of the mentioned studies dealt with square channels. However, a rectangular or triangular passage is of high importance to maintain the integrity of gas turbine blade internal cooling design. It is quite common to find rectangular cooling passages, particularly moving from the midchord to the trailing edge of the blade (or from the midchord to the leading edge of the blade); the channels must become more rectangular as the blade becomes thinner (or as the blade becomes thicker). This thinning (or thickening) of the channel changes the effective rotation-induced secondary flow pattern from that of a square duct as shown in Fig. 14. For this reason, one cannot simply apply the knowledge of the rotationally induced flow pattern and the associated surface heat transfer coefficients in a square channel to that of a rectangular channel. However, very few experimental heat transfer data on a rectangular coolant passage are available in the literature. Azad et al.[57] and Al-Hadhrami et al.[58] studied heat transfer in a two-pass rectangular rotating channel (aspect ratio equal to 2:1) with 45-deg angled ribbed walls and 45-deg V-shaped ribbed walls, respectively, including the effect of channel orientation with respect to the axis of rotation. They found that the effect of rotation on the two-pass rectangular channel is very similar to that on the two-pass square channel except the leading surface heat transfer coefficient does not vary much with the rotation compared with the square channel case. The results show that 45-deg V-shaped ribs are better than 45-deg crossed V-shaped ribs and subsequently better than 45-deg parallel angled ribs and 45-deg crossed angled ribs. The difference in heat transfer coefficients between leading and trailing surfaces is reduced when the channel has an angle with respect to the axis of rotation.

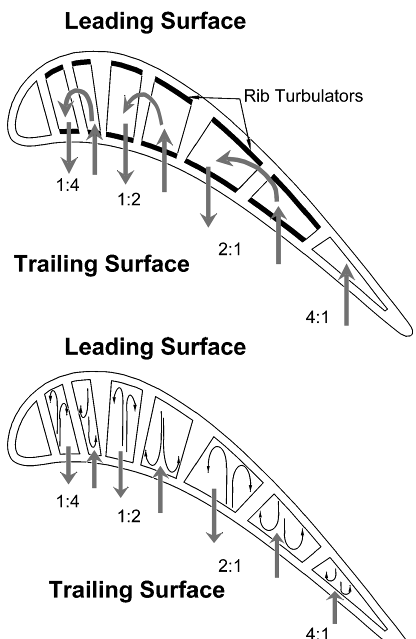


Fig. 14 Typical cooling passage size and orientation with rotation-induced secondary flow.

As the trailing edge is approached, the aspect ratio of the cooling channels continues to increase as the channel orientation is also changing. Griffith et al.[59] studied heat transfer in a single-pass rectangular channel ($AR = 4{:}1$) with smooth and 45-deg angled ribbed walls, including the effect of channel orientation with respect to the axis of rotation. Results show that the narrow rectangular ribbed channel exhibits much higher heat transfer enhancement than the square and 2:1 ribbed channels previously investigated and that there exists spanwise variations of the heat transfer coefficient in the high aspect ratio channel. Also, channel orientation significantly affects the leading and side surfaces, yet does not have much affect on the trailing surfaces for both smooth and ribbed surfaces. Lee et al.[60] also studied the heat transfer in a single-pass rectangular channel ($AR = 4{:}1$); they compared the heat transfer augmentation of parallel and staggered arrangement for V-shaped and angled rib turbulators with and without gaps, in a channel oriented at 135 deg to the axis of rotation. The results show that V-shaped rib configuration produces more heat transfer augmentation than the angled rib configuration for both the stationary and rotating cases. There is only negligible difference in heat transfer augmentation between the parallel and staggered rib configurations for both the stationary and rotating cases. The results also show that the V-shaped ribs with gaps produce overall less heat transfer augmentation that the V-shaped ribs without gaps; whereas the angled ribs with gaps produce overall greater heat transfer augmentation than the angled ribs without gaps for the stationary case, clearly the same augmentation for the rotating case. Most important, for narrow rectangular rib-turbulated channels oriented at 135 deg with respect to the plane of rotation, heat transfer augmentation on both the leading and trailing surfaces increases with rotation. This is quite different from the square channel, where rotation enhances the trailing surface heat transfer but reduces the leading surface heat transfer for the radial outward flow case. This provides positive information for the cooling designers. Wright et al.[61,62] expanded the study of the 4:1 ribbed channel to examine the overall thermal performance (not only the heat transfer augmentation, but also the pressure drop penalty). They investigated 4:1 channels with six rib configurations: angled, V-shaped, W-shaped, discrete angled, discrete V-shaped, and discrete W-shaped ribs. As shown in Fig. 15, they found that the overall thermal performance of the discrete V-shaped and discrete W-shaped ribs are better than the V-shaped and W-shaped ribs

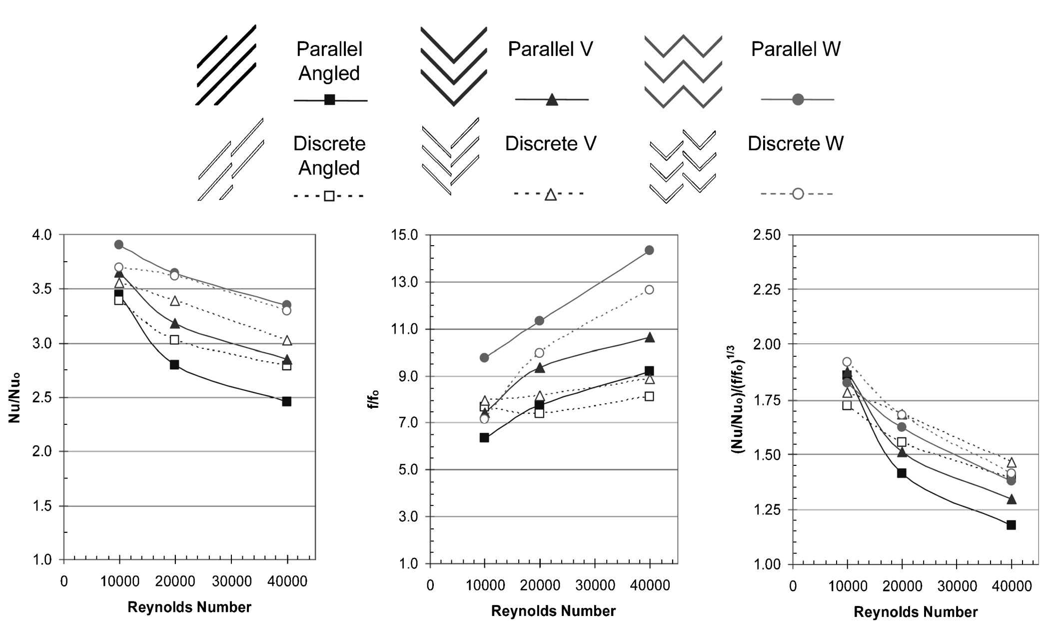


Fig. 15 Thermal performance of various rib configurations in 4:1 rotating cooling passage from Wright et al.[62]

and subsequently better than the traditional angled ribs for both stationary and rotation cases.

The cooling channels near the leading edge of the blade typically have smaller aspect ratios compared to those near the center of the blade. Fu et al.[63] investigated the heat transfer augmentation in wider aspect ratio two-pass cooling channels ($AR = 1{:}2$ and 1:4), located near the leading edge of the blade, with smooth walls and 45-deg angled ribbed walls, respectively. This study concluded the rotation effect increased heat transfer on the trailing wall but decreased the heat transfer on the leading wall in the first pass of both the 1:2 and 1:4 channels. When the results from this study were compared to the previous studies with various aspect ratios, it could be seen that the 1:4 channel has the largest heat transfer difference between the leading and trailing walls in the first pass. In the second pass, however, the difference of the heat transfer between the leading and trailing walls was dramatically reduced under the rotation condition when compared to the first pass. This implies rotation has a relatively small effect in the second pass of the 1:2 and 1:4 channels. It was suggested that the 180-deg turn-induced vortices dominate the rotation-induced vortices for the low aspect ratio channels.

With previous studies exploring a wide range of channel sizes, direct comparisons of multiple aspect ratio channels with different orientations would be beneficial for blade designers. Figure 16 summarizes the heat transfer augmentation in rotating channels with smooth walls and ribbed walls, respectively, with various aspect ratios and orientations.[63] The heat transfer enhancement is shown for one region in the first pass and one region in the second pass; these regions are sufficiently far from the entrance (first pass) and

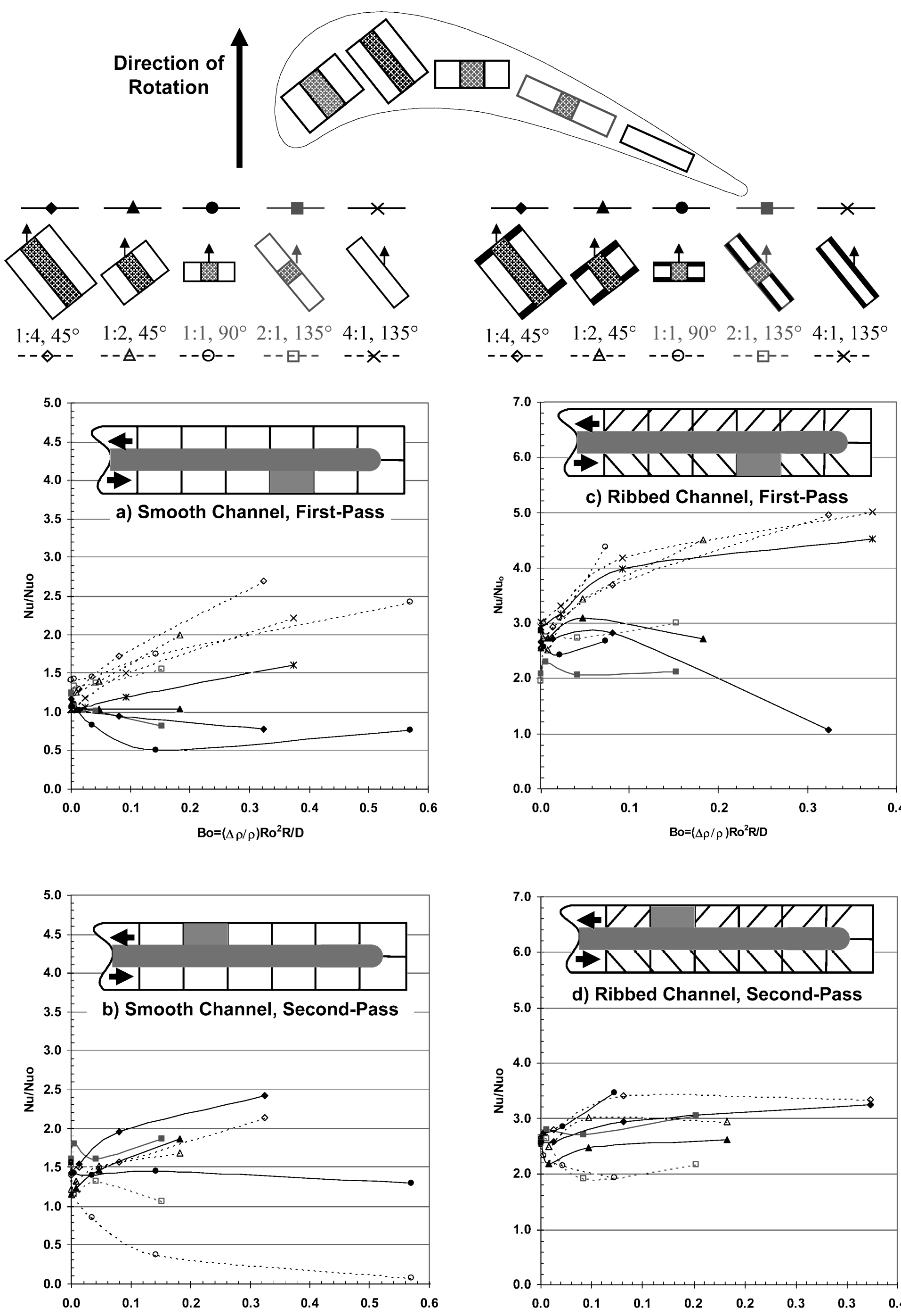


Fig. 16 Heat transfer enhancement in rotating smooth and ribbed channels from Fu et al.[63]

the turn (second pass) to represent fully developed flow for each pass. Because the orientation of the cooling passage varies to fit the cross section of a given blade, the various aspect ratio channels are shown at various orientations depending on the probable location in a turbine blade. For all aspect ratios with smooth walls, the heat transfer augmentation on the trailing surfaces in the first pass is increasing with the increasing rotating buoyancy number. However, the heat transfer decreases on the leading surface of the 2:1 and 1:4 channels, whereas it increases on the leading surface of the 4:1 channel. For the 1:1 channel, the leading-surface heat transfer decreases then increases at higher rotating buoyancy number. In the second pass of channels with smooth walls, both the leading and trailing surfaces of the 1:4 and 1:2 channels undergo heat transfer augmentation with the increasing buoyancy number, and the trailing surfaces of the 1:1 and 2:1 channels experience a decrease, whereas the leading surfaces see an increase in the Nusselt number ratios (heat transfer enhancement). The trends for the heat transfer enhancement in channels with angled ribs are less discernable due to the variation in rib geometry. The same size ribs were used in each channel; therefore, the rib height-to-hydraulic diameter (or rib height-to-channel height) ratio varies between the channels.

Computational Heat Transfer in Rotating Coolant Passages

In recent years, several researchers have made computational studies on internal cooling channels of the rotating blade. Numerical predictions provide the details that are difficult to obtain by experiments. Moreover, the increase in computation power in desktop computers has made it economical to optimize the design parameters based on numerical analyses. Most common models are based on a two-equation turbulence model, namely, the k–ε model, low Reynolds number k–ε model, the two-layer k–ε model, and the low Reynolds number k–ω model. The advanced Reynolds stress model and the second-moment closure model are also employed. In general, it has been mentioned by previous investigators, for example, Prakash and Zerkle,[64] that the k–ε model may not correctly predict the complex flow and heat transfer behaviors in rotating two-pass rectangular channels with angled rib-type turbulence promoters typically in rotor coolant passage design. This is simply due to the isotropic turbulence assumption and near-surface wall function requirement made by using the k–ε model. However, there are much more complicated flowfields in the rotor coolant passage design, such as rotating three-dimensional asymmetric flow, 180-deg turning and recirculation, angled rib-induced flow separation and secondary flow, etc. It has been found by previous investigators, for example, Lin et al.,[65] that the low Reynolds number k–ω model does predict reasonably well the flow and heat transfer in typical two-pass rotating channels with angled ribbed walls. This is attributed to the fact that turbulence dissipation is specified in terms of the turbulent kinetic energy in the near-wall region to replace wall function and very fine grid computation is used toward the surface (e.g., many papers reviewed and cited in Chapter 7 of Han et al.[1]). As expected, the second-moment closure model provides the best flow and heat transfer characteristics in typical rotating two-pass channels with angled rib turbulators. In this model, six additional transport equations are required to be solved in a three-dimensional turbulent flow and the eddy diffusivity in the momentum transport equation is replaced by the source terms developed from the turbulent Reynolds stress tensor. Therefore, more computing time is required. In the following paragraphs, only a few representative studies using the second-moment closure model presented by Chen et al.[66,67] are mentioned.

Jang et al.[68,69] predicted flow and heat transfer in a single pass rotating square channel with 45-deg angled ribs, as well as in a two-pass stationary square channel with 60-deg angled ribs by the second-moment closure model used by Chen et al.[66,67] Their[68,69] heat transfer coefficient prediction compared well with the experimental data. This has affirmed the superiority of the second-moment closure model compared to the simpler isotropic eddy viscosity k–ε model. The advantage of the second-moment closure model is that it resolves the near-wall flow all of the way to the solid wall rather than using log-law assumption in the viscous sublayer. With this near-wall closure, surface data such as heat transfer coefficients and

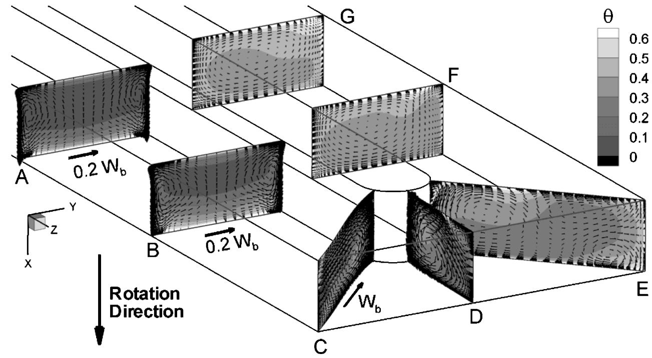
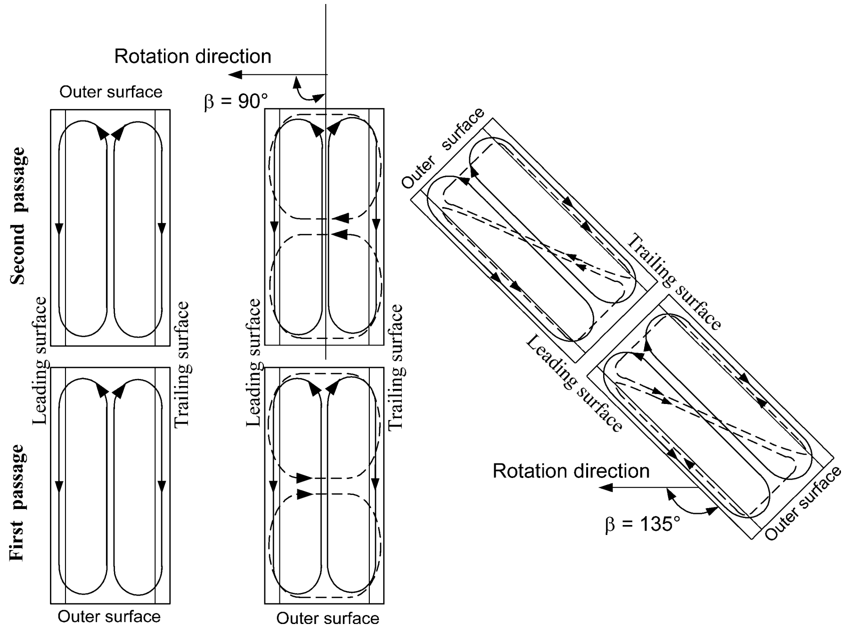


Fig. 17 Secondary flow and temperature contours in rotating 2:1 smooth channel from Al-Qahtani et al.[70]

friction factors can be evaluated directly from velocity and temperature gradient on the solid wall. Al-Qahtani et al.[70] predicted flow and heat transfer in a two-pass rotating rectangular channel (aspect ratio equal to 2:1) with smooth surface and with 45-deg angled ribs, respectively, by the same second-moment closure model used by Jang et al.[68,69] They found that the predicted heat transfer coefficient compared reasonably well with the experimental data of Azad et al.[57] Figure 17 shows the secondary flow and temperature contours in a rotating smooth channel with an aspect ratio of 2:1. Figure 18 shows the secondary flow induced by angled ribs and rotation.

Al-Qahtani et al.[71] also conducted a numerical study of flow and heat transfer in one-pass rotating rectangular channels ($AR = 4{:}1$) with 45-deg rib turbulators using the same second-moment closure model. Their heat transfer predictions were in good agreement with the experimental data of Griffith et al.[59] Using the same second-moment closure model as Jang et al.,[68,69] Su et al.[72] computed flow and heat transfer in one-pass rotating rectangular channels ($AR = 4{:}1$) with V-shaped rib turbulators for a wide range of Reynolds numbers from 10^4 to 5×10^5. Their heat transfer predictions were in good agreement with the experimental data of Lee et al.[60] Figure 19 shows a comparison of the velocity vectors and temperature contours for nonrotating rectangular channels with V-shaped ribs. Results show that for the non rotating case the V-shaped ribs induce four counter-rotating vortices that vary in size along the streamwise direction. However, for the rotating case, the rotation-induced cross-stream secondary flow distorts the V-shaped rib-induced vortices and affects the heat transfer on both leading and trailing surfaces with V-shaped ribs. Results also show that the V-shaped ribs create a symmetric heat transfer enhancement from the channel centerline toward the sidewalls, the rotation increases heat transfer enhancement on the trailing surface and decreases heat transfer enhancement on the leading surface, and higher Reynolds numbers tend to weaken the heat transfer enhancement effect of the V-shaped rib-induced secondary flow.

Su et al.[73,74] also studied the flow and heat transfer in rotating two-pass square ($AR = 1{:}1$) and low aspect ratio ($AR = 1{:}2$ and 1:4) rectangular channels with smooth walls and 45-deg angled ribs, for the rotating direction perpendicular to the channel axis. A total of 30 test cases were investigated with various combinations of Reynolds numbers ($Re = 10^4$ and 10^5), rotation numbers (0.0, 0.14, and 0.28), and coolant-to-wall density ratios (0.13, 0.20, and 0.40). For the low Reynolds number and low Ro cases, they found that the rotation effect on the Nusselt number and friction factor ratios is more significant in the $AR = 1{:}2$ channel than those observed in either the square or $AR = 1{:}4$ channel due to the presence of strong turn-induced vortices. For the high Ro and high Reynolds number cases, however, the rotation effect decreases continuously when the channel aspect ratio was changed from $AR = 1{:}1$ to 1:2 and 1:4. Figure 20 shows secondary flows in the first passage (before the 180-deg turn) of rotating ribbed channels with different aspect ratios. The preceding numerical investigations demonstrated that the advanced second-order Reynolds stress (second-moment) turbulence models are capable of providing detailed three-dimensional velocity, pressure, temperature, Reynolds stresses, and turbulent heat



a) Nonrotating, $Ro = 0.0$, and $\Delta\rho/\rho = 0.115$

b) Rotating, $Ro = 0.11$, $\Delta\rho/\rho = 0.115$, and $\beta = 90$ deg

c) Rotating, $Ro = 0.11$, $\Delta\rho/\rho = 0.115$, and $\beta = 135$ deg

Fig. 18 Secondary flow induced by angled ribs and rotation from Al-Qahtani et al.[70]: – – –, rotation induced secondary flow and ——, rib-induced secondary flow.

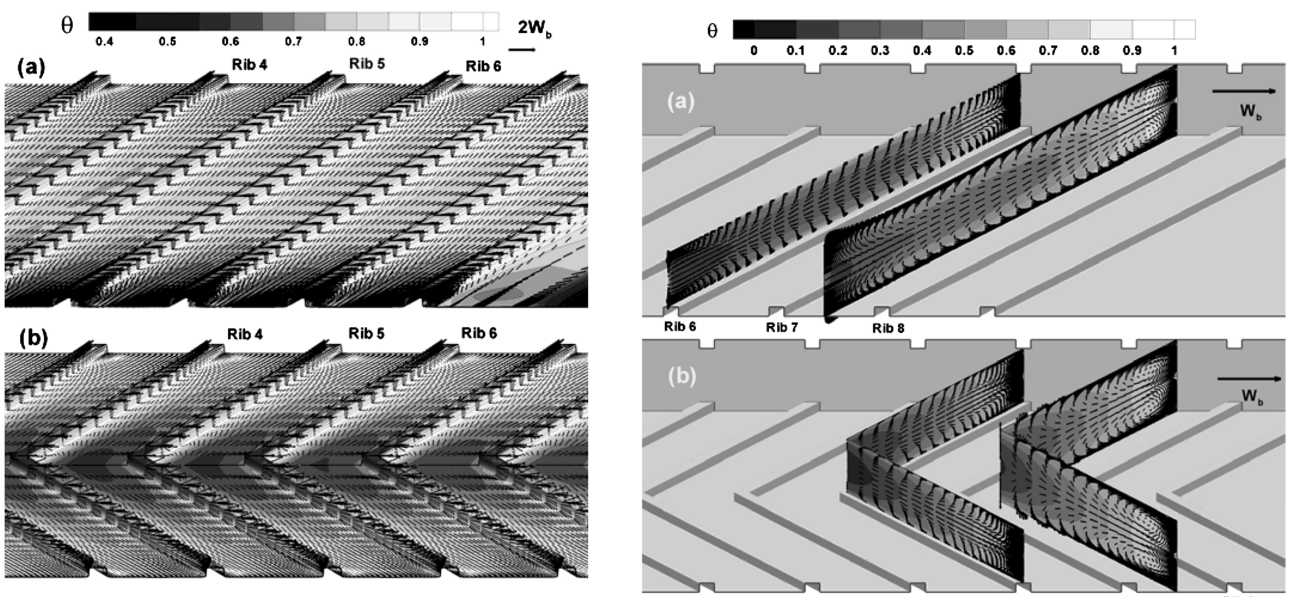


Fig. 19 Velocity vectors and dimensionless temperature contours [$\theta = (T - T_i)/(T_w\text{-}T_i)$] in nonrotating, 4:1 cooling channel with a) angled and b) V-shaped ribs from Su et al.[72]

fluxes that were not previously available in most of the experimental studies.

Rotational Effect on Jet Impingement Cooling

Jet impingement heat transfer is most suitable for the leading edge of the blade where the thermal load is highest and a thicker cross section of this portion of the blade can suitably accommodate impingement cooling. There are many studies available in open literature focused on the effects of jet-hole size and distribution, jet-to-target surface distance, spent-air crossflow, cooling-channel cross section, and the target surface shape on the target surface heat transfer coefficient distributions (e.g., many papers reviewed and cited in Chapter 4 of Han et al.[1]). However, most of impingement cooling studies are for stator blades; only a few previous studies from previous researchers focus on rotor blade impingement cooling. A group of researchers from Massachusetts Institute of Technology (Epstein et al.[75]) first studied the effect of rotation on impingement cooling in the leading edge of a rotor blade. They reported that the rotation decreases the impingement heat transfer, but the effective heat transfer is better than a smooth rotating channel. A second group of researchers, from Germany (Mattern and Hennecke[76]), reported the effect of rotation on the leading-edge impingement cooling by using the naphthalene sublimation technique. This experiment did not include the rotating buoyancy effect. The jet direction has an offset angle with respect to the rotation direction. They found that the rotation decreases the impingement heat transfer for all staggered angles. The effect of rotation is least when jet direction has an angle of 45 deg to the rotation direction. However, a maximum of 40% reduction in heat transfer is noted when the jet direction is perpendicular to the rotation direction. A third group of researchers,

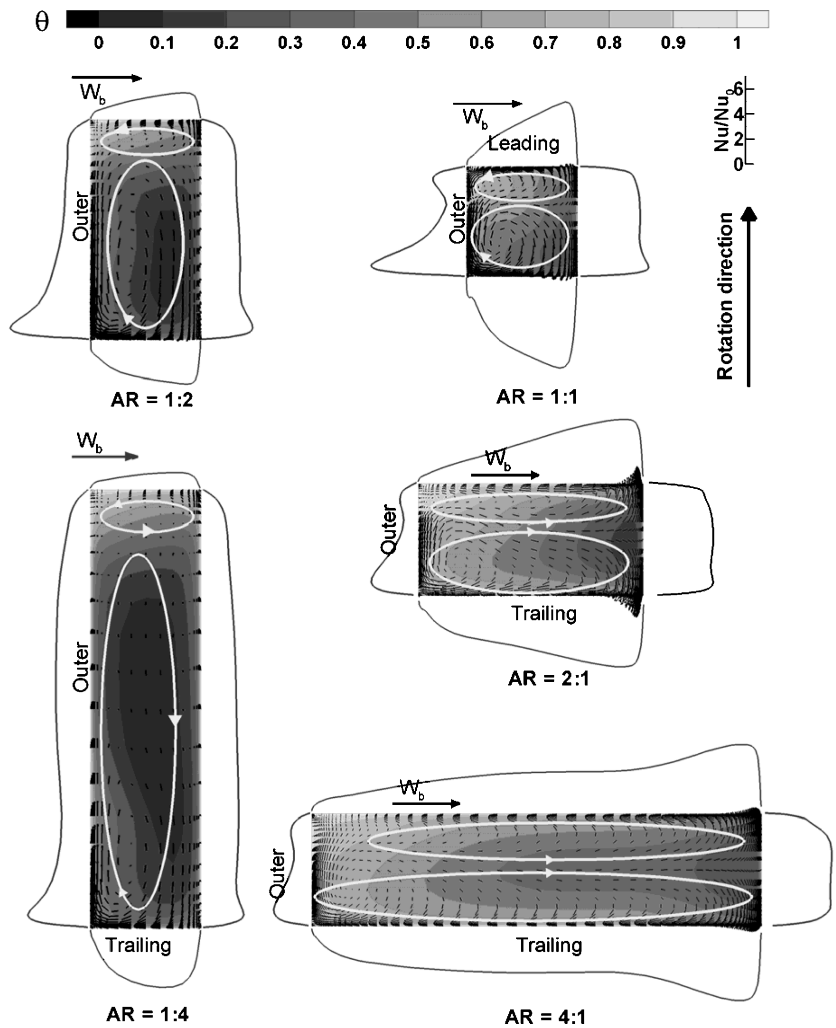


Fig. 20 Secondary flows, dimensionless temperature contours, and Nusselt number ratios in first passage (before 180-deg turn) of rotating ribbed channels with different aspect ratios from Su et al.[74]

from Solar Turbines (Glezer et al.[77]), studied the effect of rotation on swirling impingement cooling in the leading edge of a blade. They found that screw-shaped swirl cooling can significantly improve the heat transfer coefficient over a smooth channel and the improvement is not significantly dependent on the temperature ratio and rotational forces.

Parsons et al.[78,79] studied the effect of rotation on impingement cooling in the midchord region of the blade. A central chamber serves as the pressure chamber, and jets are released in either direction to impinge on two heated surfaces. The jet impinging directions have different orientations with respect to the direction of rotation. As shown in Fig. 21, they reported that the rotation decreases the impingement heat transfer on both leading and trailing surfaces with more effect on the trailing side (up to 20% heat transfer reduction). This is due to jet deflection by the Coriolis and centrifugal forces. Akella and Han[80] studied the effect of rotation on impingement cooling for a two-pass impingement channel configuration with smooth walls. The difference from the earlier experiment by Parsons et al.[78,79] is that spent jets from the trailing channel are used as cooling jets for the leading channel. Therefore, the crossflow in the trailing side is radial outward; for the leading side, it is radial inward. They[80] reported that irrespective of the direction of rotation, the heat transfer coefficient on the first-pass and second-pass impinging wall decreases up to 20% in the presence of rotation. Akella and Han[81] included 45-deg angled ribs in the target surfaces of their two-pass impingement channel with rotation. They reported that jet impingement on the ribbed wall can provide 10–50% more heat transfer compared to that on the smooth wall for jet Reynolds number increasing from 4×10^3 to 10^4. This is because the angled rib-induced secondary flow gets stronger with higher crossflow at higher jet Reynolds number. They also found that the rotation decreases impingement heat transfer on the first-pass and second-pass ribbed walls.

Rotational Effect on Pin-Fin Cooling

Pin fins are mostly used in the narrow trailing edge of a turbine blade where impingement and ribbed channels cannot be accommodated due to manufacturing constraints. Pin fins commonly used in turbine blade cooling have pin height-to-diameter ratio between 1/2 and 4. Heat transfer in turbine pin-fin cooling arrays combines the pin heat transfer and channel endwall heat transfer. Because of the turbulence enhancement caused by pins (wakes and horseshoe vortex), heat transfer from channel endwalls is higher than smooth wall cases, but casting pins will cover a considerable channel endwall area and that area needs to be compensated for by the increased pin surface area for cooling. In addition to flow disturbances, pins conduct thermal energy away from the channel endwall surface. Long pins can increase the effective heat transfer area and perform better than short pins. There have been many investigations that studied the effects of pin array (inline or staggered), pin size (length-to-diameter ratio of 0.5–4), pin distribution (stream wise and spanwise-to-diameter ratio of 2–4), pin shape (with and without a fillet at the base of the cylindrical pin, oblong-, cube-, and diamond-shaped pins, as well as the stepped-diameter cylindrical pins), partial length pins, flow convergence and turning, and with trailing edge coolant extraction on the heat transfer coefficient and friction factor distributions in pin-fin cooling channels (e.g., Metzger et al.,[82]

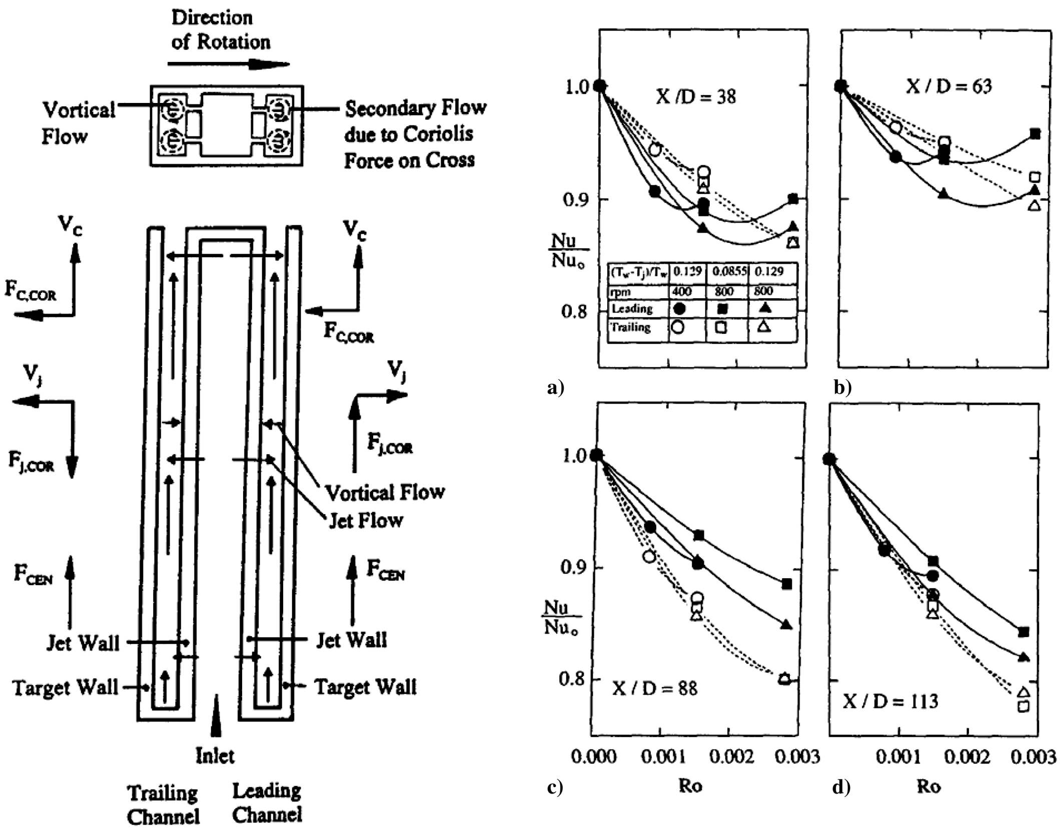


Fig. 21 Vortices and rotational forces and effect of rotation on Nusselt number ratio of target wall from Parsons et al.[78]

Chyu,[83] and many papers reviewed and cited in Chapter 4 of Han et al.[1]). However, the majority of investigations involving pin-fin cooling have been limited to stationary channels that are applicable for stator blade trailing-edge cooling designs; only a few studies focus on rotor blade pin-fin cooling.

Recently, researchers from Rensselaer Polytechnic Institute (Willett and Bergles[84]) studied effect of rotation on heat transfer in narrow rectangular channels ($AR = 10{:}1$) with smooth and with typical pin-fin array, respectively, including channel orientation effect with respect to the plane of rotation. They found that the heat transfer enhancement in the pin-fin channel due to rotation and buoyancy was less than the enhancement in the smooth channel. They showed that heat transfer enhancement mainly is due to pin-fin flow disturbance; pin fins significantly reduce the effect of rotation, but they do not eliminate the effect. Wright et al.[85] studied effect of rotation on heat transfer in narrow rectangular channels ($AR = 4{:}1$ and $8{:}1$) with typical pin-fin arrays used in turbine blade trailing-edge design and oriented at 150 deg with respect to the plane of rotation. Results show that turbulent heat transfer in a stationary pin-fin channel can be enhanced up to 3.8 times that of a smooth channel; rotation enhances the heat transferred from the pin-fin channels up to 1.5 times that of the stationary pin-fin channels.[85] Most important, for narrow rectangular pin-fin channels oriented at 150 deg with respect to the plane of rotation as shown in Fig. 22, heat transfer enhancement on both the leading and trailing surfaces increases with rotation. This provides positive information for the cooling designers.

Rotational Effect on Dimple Cooling

Dimples recently have been considered for turbine blade trailing-edge cooling designs. Dimples provide reasonable heat transfer enhancement with relatively low pressure-loss penalty as compared with the ribs and pin fins. The dimple cooling can be a good choice if the pressure loss is the main concern in the cooling design. Because of the disturbance enhancement caused by dimples, heat transfer from a dimpled surface is higher than the smooth wall conditions. This is because dimples induce flow separation and reattachment

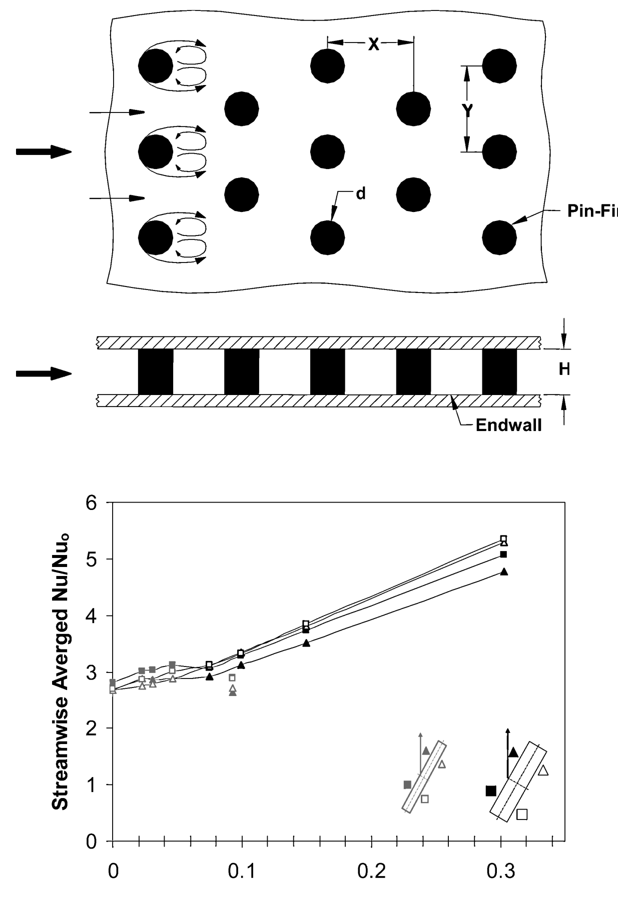


Fig. 22 Secondary flow induced by pin fins with heat transfer enhancement in rotating cooling passages with pin fins, from Wright et al.[85]

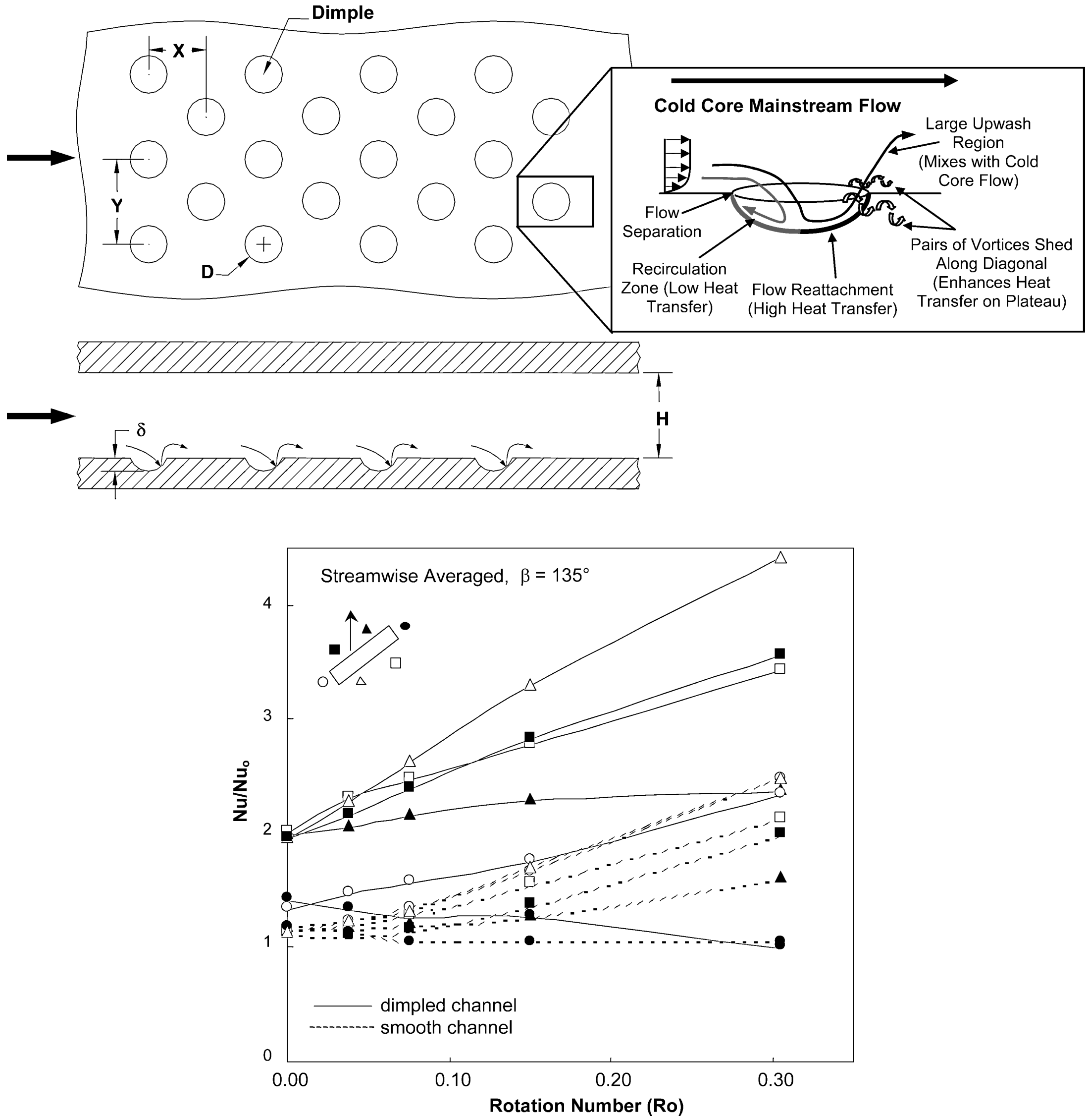


Fig. 23 Dimple-induced secondary flow and heat transfer enhancement in rotating dimpled channels, from Griffith et al.[90]

with pairs of vortices. In addition to flow disturbances, dimples increase the heat transfer area. In general, higher heat transfer enhancement occurs on the flow reattached regions either at the dimple cavity downstream or on the dimple downstream flat surface. The heat transfer enhancement is typically around 2–2.5 times that of the smooth wall value with 2–4 times pressure loss penalty and is fairly independent of Reynolds number and channel height or aspect ratio. There have been a number of previous studies (e.g., Chyu et al.,[86] Moon et al.,[87] and Mahmood et al.[88]) that evaluated the effects of dimple size, dimple depth (depth-to-print diameter ratio = 0.1–0.3), distribution, shape (cylindrical, hemispheric, and teardrop), and channel height on the heat transfer coefficient and friction factor distributions in dimple cooling channels. However, the majority of investigations involving dimple cooling have been limited to stationary channels that are applicable for stator blade trailing-edge cooling designs; only a few studies focus on rotor blade dimple cooling.

Researchers from Louisiana State University (Zhou and Acharya[89]) studied heat/mass transfer in a rotating square channel with typical dimple array. They found that heat transfer enhancement for the stationary dimple channel is around two times that of the smooth wall value; however, rotation enhances heat transfer on the trailing dimple surface and reduces heat transfer on the leading dimple surface in a similar manner as the rotational effect on the trailing and leading surfaces of the square channel with ribs. Griffith et al.[90] studied heat transfer in rotating rectangular channels (AR = 4:1) with typical dimple array on both leading and trailing walls, including the effect of channel orientation with respect to the plane of rotation. As shown in Fig. 23, they found that rotation enhances heat transfer on both trailing and leading surfaces of the narrow dimpled channel in a similar trend as the rotational effect on the trailing and leading surfaces of the narrow rectangular channel with ribs or pins; however, the heat transfer enhancement of the ribbed or pinned channel exceeds that of the dimpled channel. Also, the dimpled channel oriented at 135 deg with respect to the plane of rotation provides greater overall heat transfer enhancement than the orthogonal dimpled channel.

Summary

Han and his coworkers have systematically investigated the effects of surface heating condition, coolant passage cross section and orientation with respect to the rotation direction, and rib-turbulators' geometry (such as size, angle, shape, and distribution) on rotating channel heat transfer. They were the first to propose and validate that turbine cooling channel orientation can be redesigned so that heat transfer on both the pressure (trailing) and suction (leading) sides of the blade-cooling channel can be enhanced due to rotation. They have fully investigated the effects of various angled ribs, impinging jets, pin fins, dimples, channel aspect ratio, and orientation with respect to the rotation direction to optimize the cooling efficiency for advanced turbine rotor blade designs. They have successfully applied the Reynolds-averaged Navier–Stokes code with near-wall, second-moment closure (Reynolds stress) turbulence model, to understand the complex three-dimensional, nonisotropic turbulent flow physics and correctly predict local heat transfer coefficient distributions in rotating serpentine coolant passages with angled rib-type turbulence promoters. This implies that cooling designers may be able to predict the detailed heat transfer distributions inside rotating serpentine coolant passages instead of just relying on experimental correlations. Han and his coworkers have published more than 45 journal papers in the area of heat transfer in rotating serpentine passages (cited 33 papers in this section). These journal publications have provided references for turbine rotor blade internal cooling designs and have motivated the worldwide research interest in the new field of heat transfer in rotating channel flow.

Turbine Blade External Cooling

In turbine blade film cooling, as shown in Fig. 24, relatively cool air is injected from the inside of the blade to the outside surface, which forms a protective layer between the blade surface and hot main stream. Turbine blade film-cooling performance primarily depends on the coolant-to-hot main-stream momentum ratio (blowing ratio), temperature ratio (density ratio), film-cooling hole geometry (hole size, spacing, shape, angle from the surface, and number of rows), and location (leading edge, trailing edge, pressure and suction sides, endwall, and blade tip) under representative engine flow conditions (Reynolds number, Mach number, combustion-generated high freestream turbulence, and unsteady wake flow). In general, the higher the momentum ratio (blowing ratio), the better the film cooling protection, that is, reduced heat transfer to the turbine blade, at a given temperature ratio (density ratio), whereas the lower the temperature ratio (cooler coolant), the better the film-cooling protection at a given momentum ratio. However, too high (or too low) of momentum ratio, that is, blowing too much or too little, may reduce the film-cooling protection, because of jet penetration into the main stream (jet liftoff from the surface), or not enough coolant to cover the surface. Therefore, it is important to optimize the amount of coolant for blade film cooling at engine operating conditions. Data from numerous studies available in the open literature suggest a blowing ratio near unity is optimum, with severe penalties at either side. In the past, there have been many studies in turbine blade film cooling available in the open literature (e.g., Ito et al.,[91] Camci and Arts,[92] Nirmalan and Hylton,[93] and many papers reviewed and cited in Chapter 3 of Han et al.[1]). This paper is limited to the review a few selected papers published from Texas A&M University's Turbine Heat Transfer Laboratory.

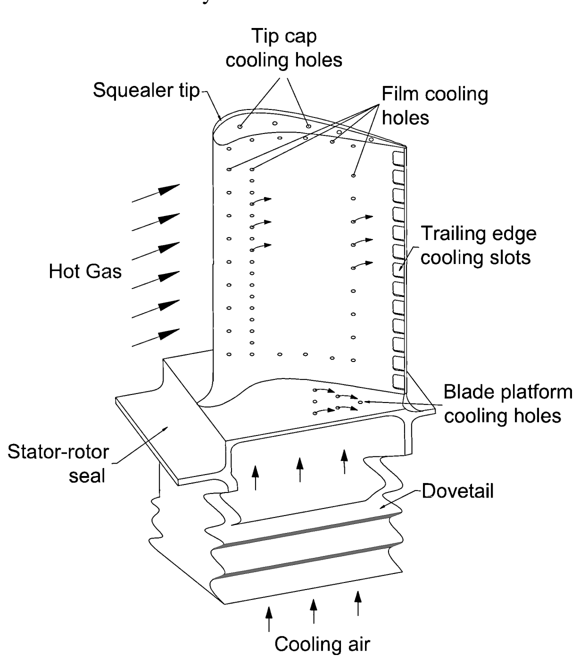


Fig. 24 Film cooling applied to external surface of turbine blade.

Turbine Blade Film Cooling

To design an effective cooling system, it is necessary to better understand the detailed hot-gas flow physics within the turbine stages. Turbine blade film cooling has been studied since the early 1970s. Since the 1980s, others have studied the effects of simulated engine flows such as unsteady, high freestream turbulence on turbine blade heat transfer penalty (e.g., Dunn et al.,[94,95] Nealy,[96] Blair et al.,[97,98] Guenette et al.,[99] Dullenkopf et al.,[100] and many previous papers reviewed and cited in Chapter 2 of Han et al.[1]). In contrast, studies of film-cooling performance under unsteady, high freestream turbulence conditions with turbulence intensities up to 15–20% were not generally available until the late 1980s and the early 1990s. Han and his coworkers were one of several research groups that explored unsteady high-turbulence effects on turbine blade film-cooling performance. Recent studies focus on combustion-generated high-turbulence effect on turbine stator heat transfer with or without film-cooling, upstream unsteady wake effect on downstream rotor blade heat transfer with or without film cooling, and surface roughness or TBCs and spallations effect on blade heat transfer with or without film cooling. Recent studies also focus on providing highly detailed heat transfer coefficient and film-cooling effectiveness distributions on turbine blades under unsteady high-turbulence flow conditions by using the transient liquid crystal image method. One important finding from these studies is that unsteady high turbulent flows do not dramatically change heat transfer coefficient distributions on the film-cooled blade, but significantly reduce the film-cooling effectiveness. This finding is crucial because the turbine blade might not be well protected by film cooling under the hostile turbomachinery flow environments. To optimize the film-cooling performance, the effects of film-hole size, length, spacing, shape, and orientation on turbine blade surface heat transfer distributions need to be considered. Results show that the shaped film-cooling hole provides better film-cooling performance than the standard cylindrical film-cooling hole.

High Freestream Turbulence Effect on Turbine Blade Leading-Edge Film Cooling

The leading edge of a blade has the highest heat transfer level and affects the heat transfer and aerodynamics over the entire blade. Thus, it would be very important to investigate the phenomena of film injection on the leading-edge region. Several investigators simulated the airfoil leading-edge film cooling using a cylinder with several film-cooling rows (e.g., Luckey et al.,[101] Mick and Mayle,[102] and a few papers reviewed and cited in Chapter 3 of Han et al.[1]). Mehendale et al.[103] and Mehendale and Han[104] investigated the effect of freestream turbulence on heat transfer and film-cooling effectiveness using a cylindrical leading-edge model. They showed that the coolant jet structures are maintained over a longer distance at lower freestream turbulence. Ou and Han[105] presented the effect of film-hole location and inclined film slots, 30 deg, on the leading-edge film-cooling heat transfer using a cylindrical leading-edge model. They showed that two-row injection (doubled total coolant flow) performs better than one-row injection at a lower blowing ratio. At higher blowing ratio, the film effectiveness of two-row injection overlaps with the effectiveness of one-row injection. Mehendale and Han[106] studied the Reynolds number effect on leading-edge film effectiveness and heat transfer coefficients. Their result showed that an increase in Reynolds number increases both the heat transfer coefficients and film-cooling effectiveness because the momentum of the flow near the wall is higher and causes more deflection and entrapment. They also reported that higher Reynolds numbers achieve lower heat flux ratios because of less film-cooling disturbance. Ekkad et al.[107] presented the effect of coolant density and freestream turbulence on a cylindrical leading-edge model using a transient liquid crystal technique to obtain the detailed heat transfer coefficient (Fig. 25) and film effectiveness distributions (Fig. 26) for the values of coolant-to-main-stream blowing ratios from 0.4 to 1.2. They also used CO_2 and air as coolant gases to simulate the density ratio effect. They showed that the film-cooling effectiveness values for air as the coolant are highest at a low blowing ratio of 0.4 and decrease with an increase in blowing ratio up to 1.2.

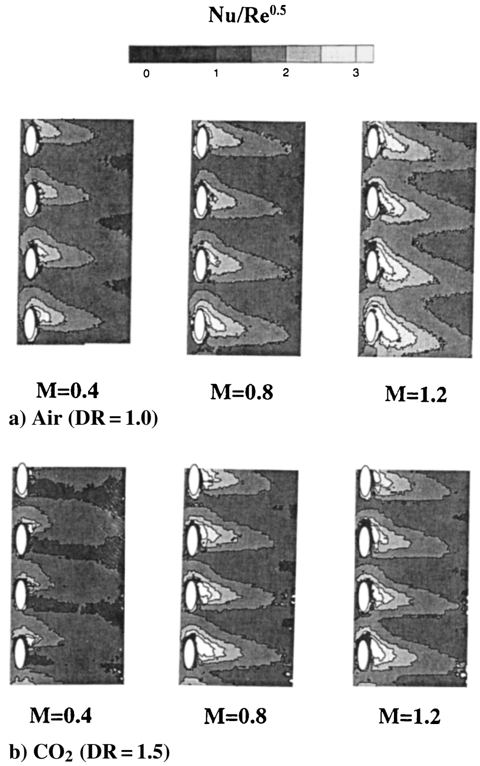


Fig. 25 Effect of blowing ratio on detailed Nusselt number distributions for air and CO_2 injection, from Ekkad et al.[107]

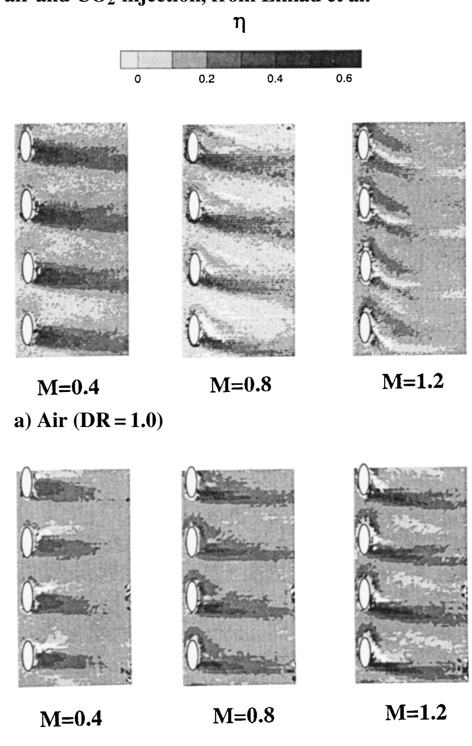


Fig. 26 Effect of blowing ratio on detailed film effectiveness distributions for air and CO_2 injection, from Ekkad et al.[107]

In the meantime, for CO_2 as the coolant, the highest film-cooling effectiveness is obtained at a blowing ratio of 0.8.

Unsteady Wake Effect on Turbine Blade Film Cooling Using Transient Liquid Crystal Technique

It is well known that downstream blades are affected by unsteady wakes shed by upstream vane trailing edges. Ou et al.,[108] Mehedale et al.,[109] Ou and Han,[110] Mehendale et al.,[111] Jiang et al.,[112] and Ekkad et al.[113] used the rotating spoked wheel-type wake generators,

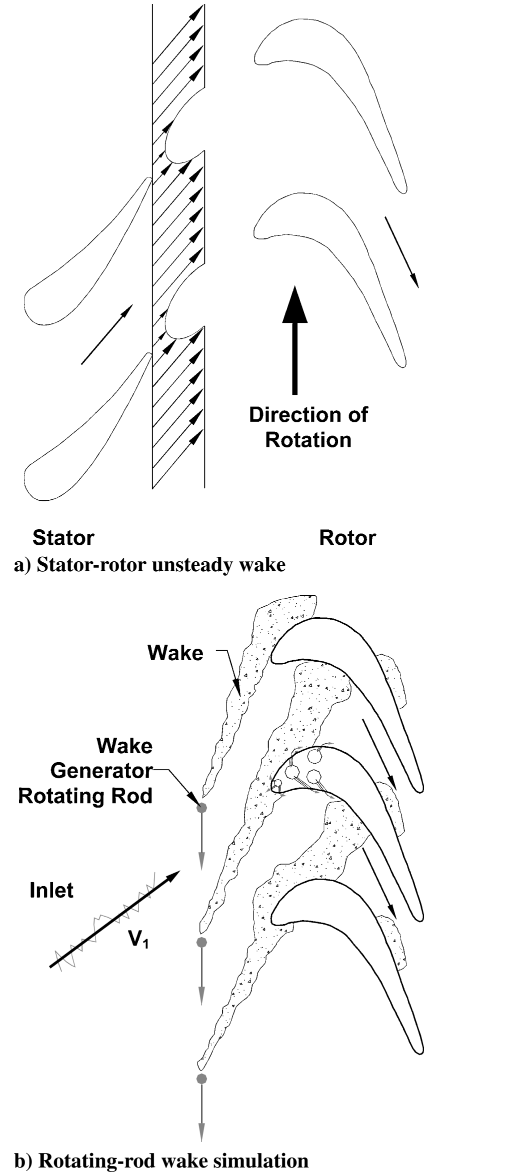


Fig. 27 Actual stator rotor interaction and laboratory simulation to study unsteady wake effect, from Mehandale et al.[109]

as shown in Fig. 27, to simulate the effect of unsteady wake on downstream stationary blade film-cooling performance. A typical turbine blade was placed in a large-scale low-speed linear two-dimensional turbine cascade facility. The test blade has three rows of film-cooling holes on the leading edge and two rows each on the pressure and suction surfaces. Air and CO_2 were used as coolants to simulate different coolant-to-main-stream density ratio effect for a range of coolant-to-main-stream blowing ratio. Du et al.[114–116] used a transient liquid crystal technique to measure the detailed heat transfer coefficient and film-cooling effectiveness distributions on the same film-cooled blade from the same unsteady wake simulation facility. They concluded that heat transfer coefficients for a film-cooled blade are much higher compared to a blade without film injection. In particular, film injection causes earlier boundary-layer transition on the suction surface of the blade. As shown in Fig. 28, unsteady wakes only slightly enhance heat transfer coefficients but significantly reduce film-cooling effectiveness on a film-cooled blade suction surface compared with a film-cooled blade without unsteady wakes. This is because the heat transfer coefficients on a film-cooled blade are already very high due to high turbulence and mixing caused by jet interaction with the main stream. The additional mixing caused by unsteady wakes might not increase much more on already highly disturbed flow heat transfer coefficients. However, the additional mixing caused by unsteady wakes would reduce protection of the surface by the coolant jets.

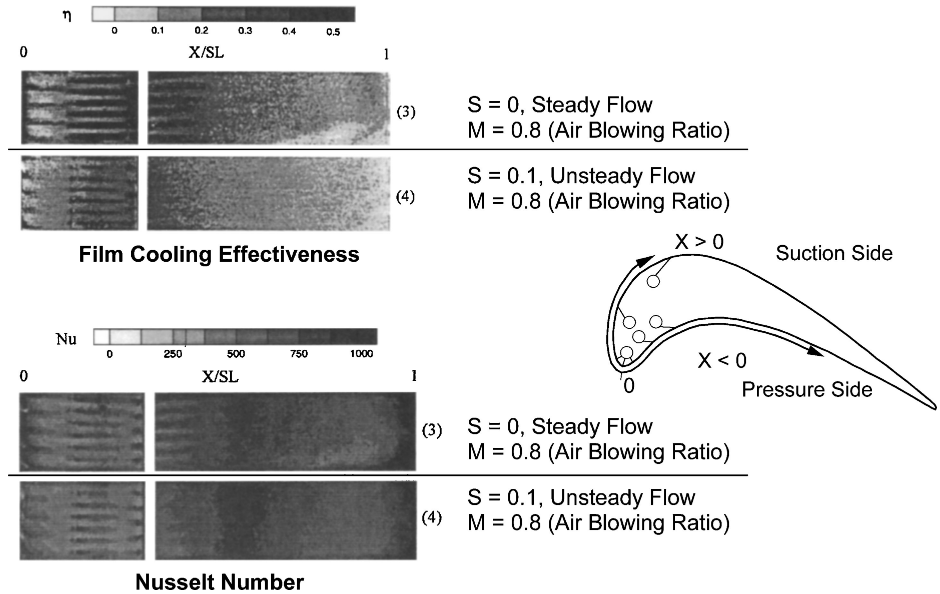


Fig. 28 Unsteady wake effect on film-cooling effectiveness and Nusselt number distributions on suction surface, from Du et al.[115]

Film Hole Shape Effect on Turbine Blade Film Cooling

To improve cooling effectiveness, one solution is to contour the film-hole geometry. Film-cooling holes with a diffuser-shaped expansion at the exit portion of the holes are believed to improve the film-cooling performance on a gas turbine blade. The increased cross-sectional area at the film-hole exit compared to a standard cylindrical hole leads to the reduction of the coolant jet velocity for a given blowing ratio. The momentum flux of the jet exiting the shaped hole and the penetration of the jet into the main stream will be reduced, which results in an increased film-cooling effectiveness. Furthermore, lateral expansion of the hole provides an improved lateral spreading of the jet, which leads to a better lateral film-cooling coverage of the blade. A few previous studies have shown that expanding the exit of the cooling hole improves film-cooling performance compared to a standard cylindrical hole (e.g., Schmidt et al.,[117] Gritsch et al.,[118] and several papers reviewed and cited in Chapter 3 of Han et al.[1]). Teng et al.[119,120] studied the effect of film-hole shape on turbine blade film-cooling performance under unsteady wake flow conditions. They used the same unsteady wake simulation facility as Du et al.[114–116] and focused on only one row of film holes near the suction-side gill-hole region to investigate the hole-shape effect on the curved blade surface under strong flow acceleration conditions. Film-cooling holes with and without exit expansions are tested and compared under steady and unsteady wake flow conditions. Detailed heat transfer coefficient and film-cooling effectiveness distributions downstream of the injection were measured by using a transient liquid crystal image technique. They found that both fan-shaped and laidback-fan-shaped holes have much lower heat transfer coefficients right after the film injection location when compared with cylindrical holes under the same unsteady wake flow conditions. They have almost the same boundary-layer transition location as the cylindrical film-hole case, but their heat transfer coefficients are higher after transition into the turbulent region. In general, as shown in Fig. 29, fan-shaped holes provide better film-cooling effectiveness than laidback-fan-shaped holes and, consequently, much better film-cooling effectiveness than cylindrical holes. Both fan-shaped holes and laidback-fan-shaped holes provide lower spanwise averaged heat flux ratio and, thus, better thermal protection over the blade surface, especially under unsteady wake flow conditions.

TBC Spallation Effect on Film Cooling

TBCs are often used to protect turbine component metal surfaces from high-temperature gases. The metal surface is coated by spraying a thin layer of high-temperature resistant TBC material. Power-generation turbines typically use natural gasses or coal-derived fuels to economize the power production. With the use of these fuels, the TBC surface may undergo severe thermal stresses, high-temperature, erosion, and corrosion during the turbine operation. These stresses may potentially cause the TBC layer to peel and expose the inner metal surface. The peeling of the TBC layer due to exposure to extremely harsh environments may produce a spallation. Spallation leads to a loss of thermal protection of the inner metal surface and enhancement of the heat transfer coefficients. The enhanced heat transfer coefficients increase the heat loads around the spallation. This could result in a rapid failure of the exposed turbine components and penalize the overall efficiency and life of the entire gas turbine engine. The spallation can occur at random, and that there is no defined shape or size of the spall make it difficult to analyze the actual spallation phenomena occurring on a real turbine blade. Thus, it needs to be modeled with predefined shape, size, and location to understand its effect on local heat transfer coefficients and film cooling effectiveness. However, the literature lacks information on the effect of TBC spallation on turbine blade film cooling and heat transfer.

Ekkad and Han[121] studied the effect of simulated TBC spallation shape, size, and depth on heat transfer enhancement over a flat surface. This study correlated the effects of various geometrical parameters of such simulated spallation cavities on heat transfer. They presented detailed heat transfer enhancement distributions using a transient liquid crystal technique. They found that the spallation can enhance the local heat transfer coefficients up to two times as compared to that with the smooth surface. Ekkad and Han[122] studied the effect of simulated TBC spallation on a cylindrical leading-edge model of a non-film-cooled blade. Detailed heat transfer coefficient distributions were obtained in their studies using a transient liquid crystal imaging method. They reported that, as shown in Fig. 30, the surface heat transfer coefficients can be enhanced up to two times in the presence of spallation compared to that for the surface without film cooling and spallation. The depth of the spallation also has a strong effect on the local heat transfer coefficients. They further reported that the freestream turbulence increases the local heat transfer coefficients over the entire leading-edge surface. Ekkad and Han[123] studied the detailed heat transfer coefficient and film-cooling effectiveness distributions on a cylindrical leading-edge model with simulated TBC spallation using a transient liquid crystal technique. The two rows of film cooling holes located at +15 and −15 deg from stagnation. The simulated spallation cavities were rectangular in shape and had rounded edges and are similar to the spallations that typically occur on the turbine blade. The simulated spallation was placed at two locations, 20–40 deg and 35–45 deg from stagnation, respectively. Results show that the heat transfer coefficients increase and film effectiveness values decrease with an increasing blowing

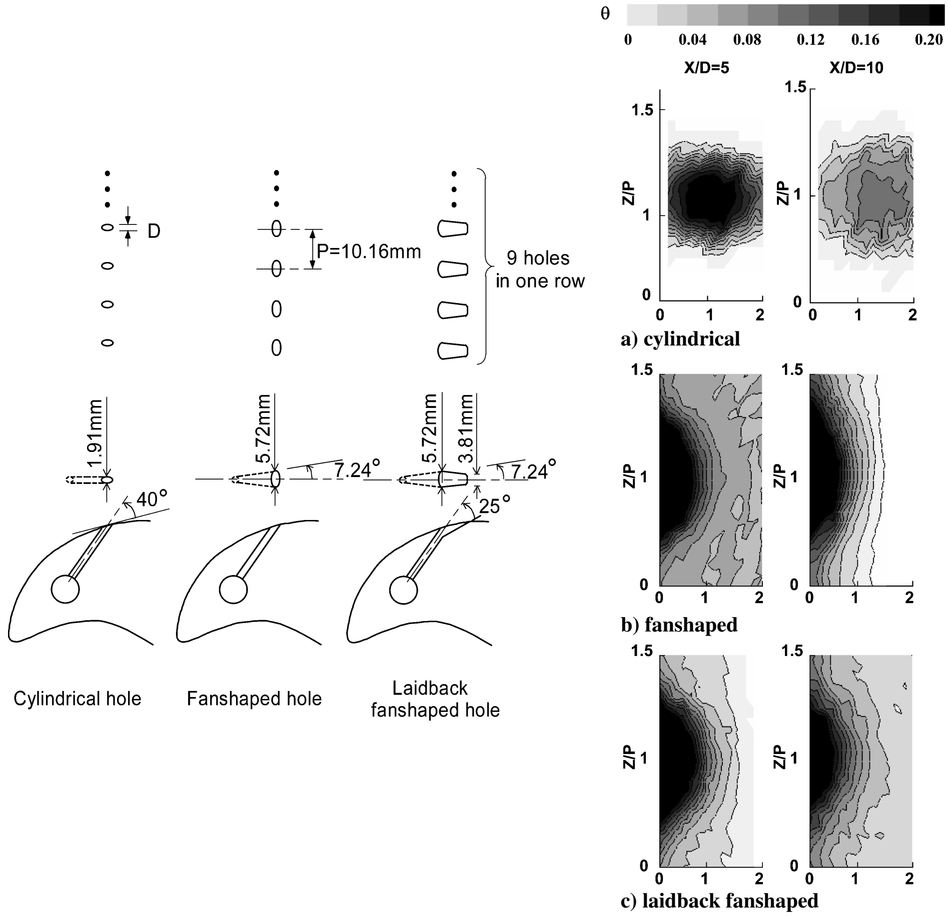


Fig. 29 Comparison of various film-hole geometries and corresponding dimensionless temperature Contours inside film-cooling boundary layers, from Teng et al.[120]

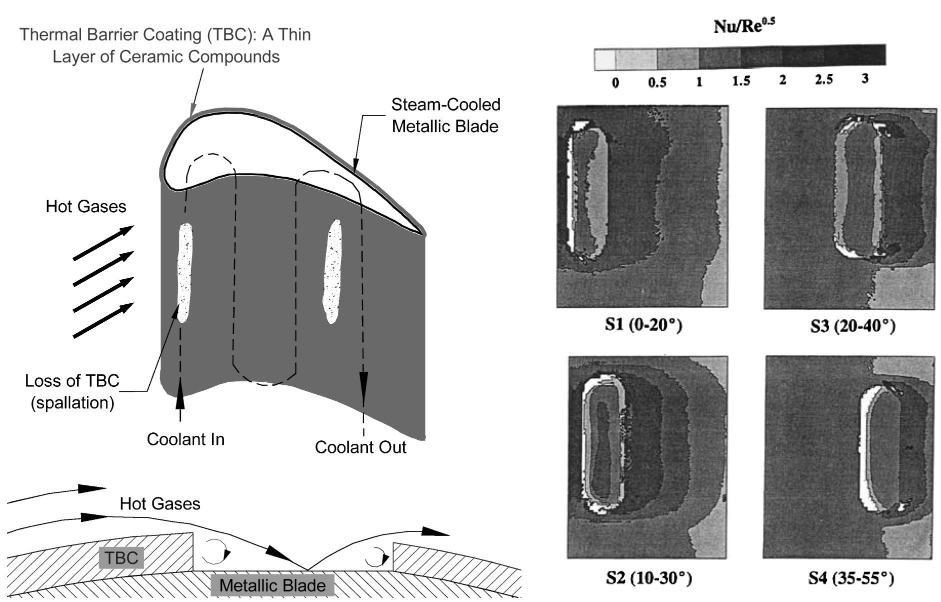


Fig. 30 Effect of TBC spallation on blade surface heat transfer, from Ekkad and Han.[122]

ratio. An increase in freestream turbulence has a very little effect on heat transfer coefficients but reduces the film-cooling effectiveness significantly at lower blowing ratios. In general, presence of spallation enhances heat transfer coefficients and causes variation in film-cooling effectiveness distributions.

Turbine Blade Tip Heat Transfer

Turbine blade tip and near-tip regions are typically difficult to cool and are subjected to potential damage due to high thermal loads. Blade tips at high temperatures may wear out due to hard rubs against the shroud. Unshrouded blades have a gap existing between the blade tip and the shroud surface, which is known as the tip gap. The leakage flow accelerates due to a pressure difference between both the pressure and suction sides of the blade, causing thin boundary layers and high heat transfer rates. This tip leakage flow is undesirable because it creates extremely high heating at the pressure side tip corner from midchord to trailing edge. As the blade tip is oxidized away, the tip gap width increases, allowing more leakage flow through the tip gap and accelerating blade tip failure. Thus, it increases the losses of turbine efficiency. It has been recognized that the blade tip geometry and subsequent tip leakage flows have a significant effect on the aerodynamic efficiency of turbines. The influence of tip gap on turbine efficiency is so significant that designers have a strong desire to improve the efficiency by decreasing the tip-to-shroud operating gaps, or by implementing more effective tip clearance controls. However, it is difficult to seal the hot-leakage flow through the tip gap completely. A common technique to reduce the tip leakage flow is to use a recessed tip, which is known as a squealer tip. A squealer tip allows a smaller tip clearance, without the risk of a catastrophic failure, in case the tip rubs against the shroud during turbine operation. The tip recess also acts as a labyrinth seal to increase flow resistance and reduce the tip leakage flow. Thus, it is important to understand both the flow and heat transfer behavior on the squealer tip of a gas turbine blade. Reliable experimental data are also important to develop and validate computational codes to predict flow and heat transfer distributions on turbine blades.

There are some papers available in open literature that discuss heat transfer coefficients on the blade tip and near-tip regions (e.g., a few papers reviewed and cited in Chapter 2 of Han et al.[1]). Several of these papers present results under engine representative mainstream flow conditions (Metzger et al.,[124] Ameri et al.,[125] Bunker et al.,[126] and Dunn and Haldeman[127]). Azad et al.[128,129] used transient a liquid crystal technique to study heat transfer coefficient and pressure distributions on a gas turbine blade tip in a five-bladed stationary linear cascade in a blowdown facility. The blade is a two-dimensional model of a first stage GE-E^3 gas turbine rotor blade. The freestream Reynolds number, based on the axial chord length and the exit velocity, was 1.1×10^6, and the inlet and the exit Mach numbers were 0.25 and 0.59, respectively. Turbulence intensity level at the cascade inlet was 9.7%. All measurements are made at three different tip gap clearances of about 1, 1.5, and 2.5% of the blade span. Results show various regions of high and low heat transfer coefficient on the tip surface, tip clearance has a significant influence on local tip heat transfer coefficient distribution, and heat transfer coefficient increases about 15–20% along the leakage flow path at higher freestream turbulence intensity level of 9.7% over 6.1%. They compared squealer-tip and plane-tip geometry and concluded that the overall heat transfer coefficients were lower for the squealer-tip case. Azad et al.[130] and Kwak et al.[131,132] investigated the heat transfer on several different squealer geometries. They found that a suction-side squealer tip gave the lowest heat transfer coefficient among all cases studied, as shown in Figs. 31 and 32. Heat transfer coefficient distributions for plane, squealer-tip and near-tip regions were presented by Kwak and Han.[133,134] When a squealer tip was used, heat transfer coefficient was found to decrease on the tip and near-tip regions.

Turbine Blade Tip Film Cooling Using Transient Liquid Crystal Technique

Experimental investigations performed in the general area of film cooling on a blade tip are limited with few papers available in open literature.[135] Figure 33 shows turbine blade-tip film cooling for both the plane and squealer-tip profiles. Kwak and Han[136] studied the local heat transfer coefficient distribution and film-cooling effectiveness using hue-detection-based transient liquid crystal technique on the blade tip for plane tip geometry. A GE-E^3, five-blade linear cascade was used similar to the one used in the previous papers by Azad et al.[128,129] The freestream Reynolds number, based on the axial chord length and the exit velocity, was 1.1×10^6, and the inlet and the exit Mach numbers were 0.25 and 0.59, respectively. Turbulence intensity level at the cascade inlet was 9.7%. All measurements were made at three different tip gap clearances of 1, 1.5, and 2.5, of blade span. The blade model was equipped with a single row of film-cooling holes at both the tip portion along the camber line and near the tip region of the pressure side. They used the three tip gap clearances (1.0, 1.5, and 2.5% of blade span) along with three average blowing ratios (0.5, 1.0, and 2.0) for the coolant. Results showed that, in general, heat transfer coefficient and film-cooling effectiveness increased with increasing tip-gap clearance. As blowing ratio increased, heat transfer coefficient decreased, while film-cooling effectiveness increased. Results also showed that adding pressure-side

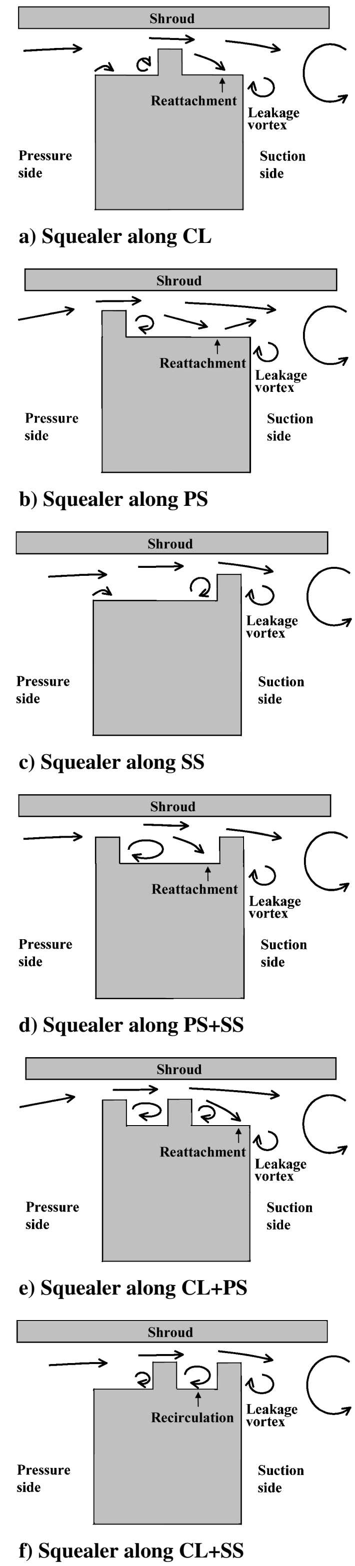


Fig. 31 Squealer-tip configurations with leakage flow in tip region, from Kwak et al.[131]

coolant injection would further decrease blade-tip heat transfer coefficient but increase film-cooling effectiveness. Kwak and Han[137] also reported the local heat transfer coefficient and film-cooling effectiveness using hue-detection-based transient liquid crystal technique on the blade tip for squealer-tip geometry. The test blade had a squealer (recessed) tip with a 4.22% recess. The blade model was equipped with a single row of film-cooling holes on the pressure-side near-tip region and the tip surface along the camber line. Results showed that the overall heat transfer coefficients increased with

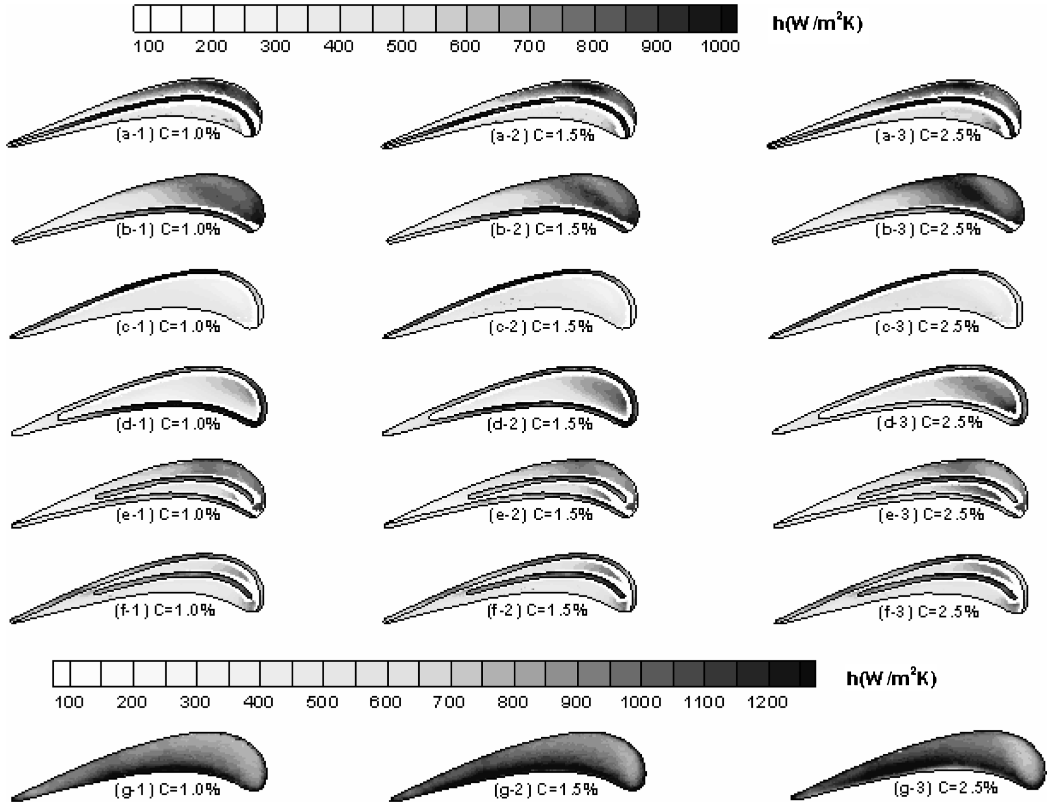


Fig. 32 Heat transfer coefficient on blade tip with various tip clearances, $C = 1$, 1.5, and 2.5, for a) squealer along centerline, b) squealer along PS, c) squealer along SS, d) squealers along both PS and SS, e) squealers along both centerline and PS, f) squealers along both centerline and SS, and g) plane tip blade, from Kwak et al.[131]

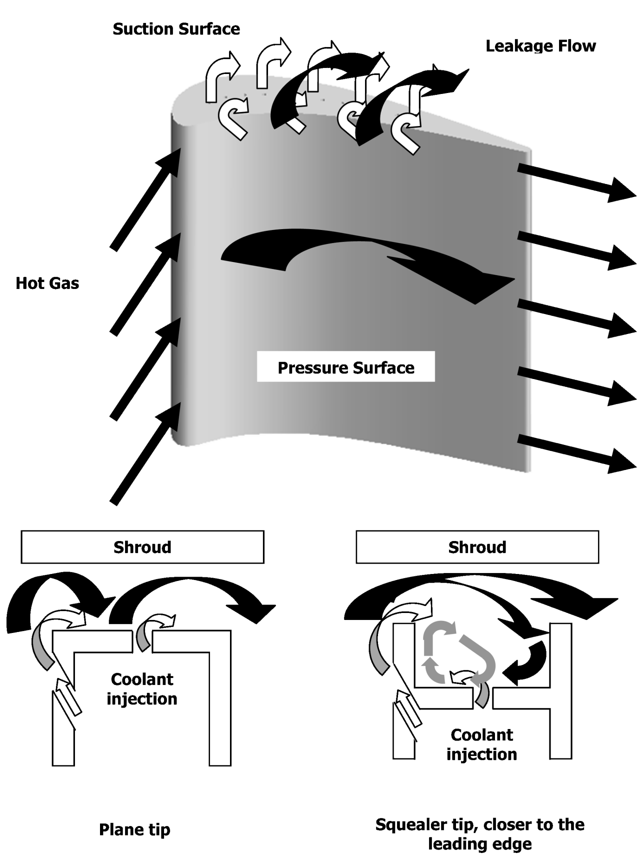


Fig. 33 Blade tip leakage flow on both plane- and squealer-tip profiles with film cooling.

increasing tip-gap clearance, but decreased with increasing blowing ratio. However, the overall film-cooling effectiveness increased with increasing blowing ratio. Results also showed that the overall film-cooling effectiveness increased but heat transfer coefficients decreased for the squealer tip when compared to the plane tip at the same tip-gap clearance and blowing ratio conditions.

Turbine Blade Tip Film Cooling Using Pressure Sensitive Paint Technique

The film-cooling effectiveness distributions were measured on the blade tip using pressure sensitive paint (PSP) technique. The motivation was to do a parametric investigation on the effect of blowing ratio, tip-gap clearance, and tip geometry on the pressure and the film-cooling effectiveness on the blade tip for plane, as well as squealer, geometry. Although the hue-detection-based transient liquid crystal technique has been used by Kwak and Han[136,137] to study the detailed local film-cooling effectiveness, it is hindered by conduction effects near sharp edges such as a film-cooling hole resulting in relatively large errors in that region. Moreover, they used a blowdown facility, where the flow needed some time (2.5 s) to reach the expected steady value, and during that developing time, the unavoidable main-stream initial flow affected the blade tip initial temperature. Considering the high heat transfer coefficient and short experiment time, the error from the initial developing time can affect blade tip heat transfer and film-cooling effectiveness. However, the PSP technique is based on mass transfer analogy, no heating of the test section or coolant is required, and the tests are performed under steady flow conditions. Thus, conduction errors at the edges and initial temperature errors are avoided.

Zhang et al.[138] were the first research group using the PSP technique in a turbine nozzle film-cooling study. Ahn et al.[139] presented film-cooling effectiveness results using the PSP technique on a plane and squealer blade tip with one row of holes on the camber line and another row of angled holes near the pressure-side tip. They used the same high flow cascade as the previous study by Kwak and Han[136,137] and investigated the effects of tip-gap clearance (1.0, 1.5, and 2.5 blade span) and blowing ratio ($M = 0.5$, 1, and 2). Air and

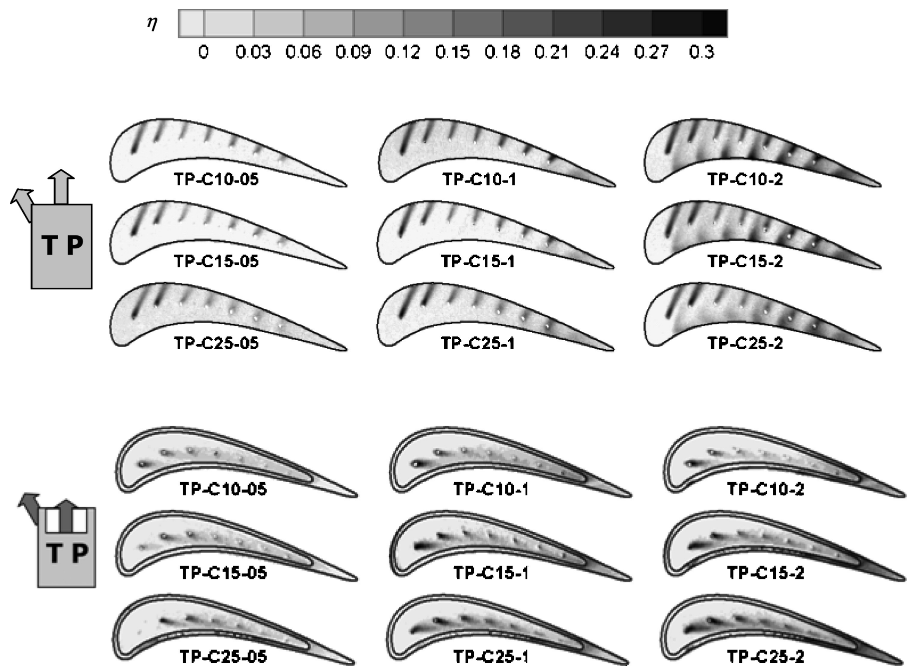


Fig. 34 Film-cooling effectiveness on plane-tip and squealer-tip blades with film holes on both tip and PS rim (TP) for various tip clearances, $C = 1$, 1.5, and 2.5%, and various blowing ratios 0.5, 1.0, and 2.0, from Ahn et al.[139]

nitrogen gas were used as the film-cooling gases, and the oxygen concentration distribution for each case was measured. The film-cooling effectiveness information was obtained from the difference of the oxygen concentration between air and nitrogen gas cases by applying the mass transfer analogy. The average blowing ratio of the cooling gas was 0.5, 1.0, and 2.0. Tests were conducted with a stationary, five-bladed linear cascade in a blowdown facility. The freestream Reynolds number, based on the axial chord length and the exit velocity, was 1.1×10^6, and the inlet and the exit Mach numbers were 0.25 and 0.59, respectively. Turbulence intensity level at the cascade inlet was 9.7%. Figure 34 shows the film-cooling effectiveness distributions on blades with a squealer-tip profile. Results show that the locations of the film-cooling holes and the presence of squealer have significant effects on surface static pressure and film-cooling effectiveness. They noted that higher blowing ratios give higher effectiveness. Results with plane tip showed clear traces of the coolant path, whereas for squealer tip, coolant accumulation effects were observed on the cavity floor.

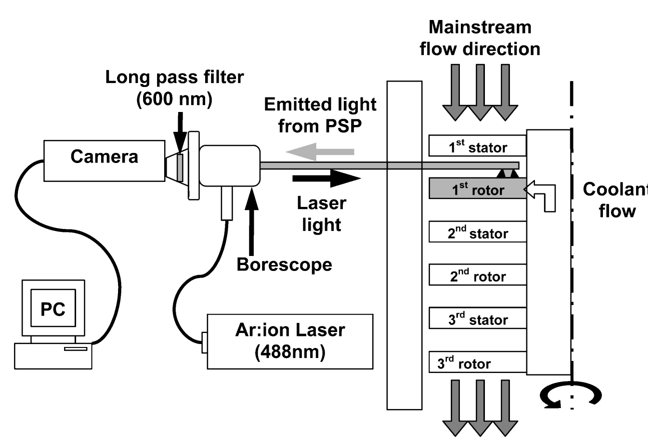


Fig. 35 Schematic of optical components used for PSP measurements on leading edge of rotating blade, from Ahn et al.[143]

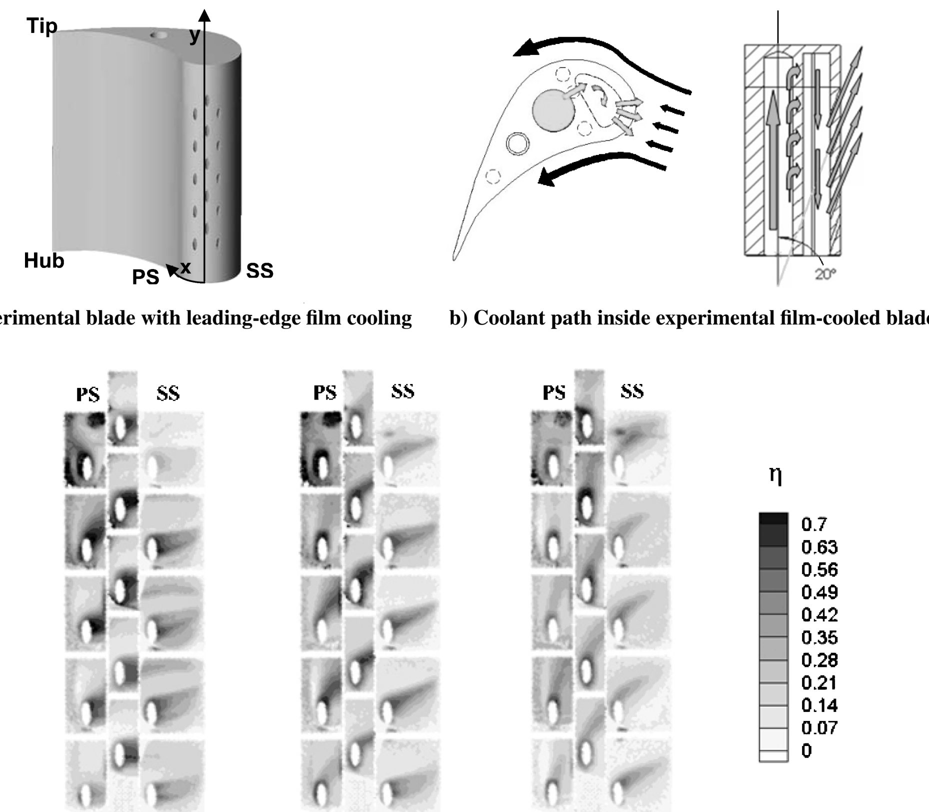
Effect of Rotation on Turbine Blade Film Cooling

Rotational Effect on Turbine Blade Film Cooling

Despite of the numerous studies on the film-cooling effectiveness, most of them were simulated in a cascade or in a wind tunnel using a flat plate or a cylindrical model. Only few results about film-cooling effectiveness under rotating engine conditions are available in open literature due to the difficulty in conducting experiments. Most of the published film-cooling study papers are based on the nonrotating test conditions. However, very limited numbers of rotating blade film-cooling study have been reported (e.g., only three papers reviewed and cited in Chapter 3 of Han et al.[1]). From these previous studies (Dring et al.,[140] Takeishi et al.,[141] and Abhari and Epsein[142]), it was concluded that film-cooling effectiveness on the rotating blades is very different from that on stationary blades, particularly, those on the pressure surface of the airfoil due to rotation-induced radial component of the film coolant trajectory. The rotation-induced radial component of the film coolant trajectory depends on film-hole location and geometry (inclined angle), as well as coolant-to-mainstream velocity and density ratios and the rotation speed. Therefore, the effect of rotation on the leading-edge region will be different from that on the pressure side (PS) and suction side (SS) of the blade, the PS near-tip film cooling will be different from the blade midspan region, and the cylindrical film holes will be different from the shaped film holes. However, there are no systematic investigations on the effects of rotation and film-hole geometry and location on rotating film-cooling performance. Most of the turbine blade-cooling designs are based on the nonrotating blade and very limited rotating blade data.

Rotational Effect on Turbine Blade Leading-Edge Film Cooling Using PSP Technique

Ahn et al.[143] studied the detailed film-cooling effectiveness distributions on the leading-edge region of a rotating blade applying the earlier mentioned PSP technique for the rotating turbine facility. The film-cooling effectiveness information was obtained from the oxygen concentration difference between air and nitrogen injection cases by applying the mass transfer analogy. The blowing ratio was controlled to be 0.5, 1.0, and 2.0, whereas the density ratio of about 1.0 was obtained using nitrogen as coolant gas. Tests were conducted on the first-stage rotor of a three-stage axial turbine with offdesign condition at 2400 rpm. The Reynolds number based on the axial chord length and the exit velocity was 2×10^5, and the total to exit pressure ratio was 1.12 for the first rotor. The film-cooling effectiveness distributions were presented along with the discussion on the



a) Experimental blade with leading-edge film cooling **b) Coolant path inside experimental film-cooled blade**

c) $M = 0.5$, N_2 **d) $M = 1.0$, N_2** **e) $M = 2.0$, N_2**

Fig. 36 Effect of rotation on leading-edge film-cooling effectiveness (rotation speed = 2400 rpm, offdesign condition), from Ahn et al.[143]

influence of blowing ratio and the vortices around the leading-edge region. The PSP technique could measure the detailed film-cooling effectiveness distributions under a rotating condition, as shown in Fig. 35. In general, as shown in Fig. 36, the PS row region shows higher averaged effectiveness values, whereas the SS row region is lower than the other due to the direction of the traces. As the blowing ratio increases, the film-cooling effectiveness level on the SS decreases, and that on the PS increases up to $M = 1.0$ and then decreases at $M = 2.0$. The level on the stagnation-line region was not affected much to the blowing ratio. Film traces accumulation effect shifted from the streamwise direction to the spanwise direction with increasing blowing ratio. In the nitrogen case, the $M = 0.5$ case has the highest overall averaged value and the overall averaged value decreases as the blowing ratio increases.

Summary

Han and his coworkers have systematically investigated unsteady high freestream turbulence effects on turbine blade film cooling with a special emphasis on turbine edge-region film cooling, including the simulated TBC spallation effect on film-cooling. In addition, they applied the transient liquid crystal image technique to provide detailed heat transfer coefficient and film-cooling effectiveness distributions on the film-cooled blade for the first time. This is very important because this technique allows designers to identify the best film-cooling geometry (such as film hole shape, angle, and distribution) to maximize the film-cooling efficiency and eliminate the potential hot spots in newly developed complex film-cooled blades. They further applied the newly developed PSP to provide detailed film-cooling effectiveness distributions on the blade tip region, as well as on the leading-edge region of a rotating blade. They have published more than 45 journal papers in this area (30 papers cited in this section) for the gas turbine heat transfer community.

Conclusions

For turbine blade external cooling, most available experimental data are for the main body of turbine blade heat transfer and film cooling. Recent research focuses on unsteady wake, high freestream turbulence, and surface roughness/TBC spallation effects on turbine rotor blade heat transfer with film cooling. To optimize the film-cooling performance, effects of film-hole size, length, spacing, shape, and orientation on turbine blade heat transfer distributions need to be considered. Satisfying the even higher turbine operating temperature requirement for higher power and efficiency makes turbine blade edge cooling an urgent issue for this century's new supercooled gas turbine blades. Turbine blade edge cooling and heat transfer includes turbine blade leading edge, trailing edge, tip and platform, with and without film cooling, under engine Mach and Reynolds number flow conditions. The effect of rotation on turbine blade film cooling and heat transfer must be addressed. Highly accurate and highly detailed local heat transfer and film-cooling data in turbine blade main body as well as turbine blade edge regions would be critical in preventing blade failure due to local hot spots. Flow visualizations, measurements, and computational fluid dynamics (CFD) predictions would provide valuable information for designing effective cooled blade for advanced gas turbines.

For turbine blade internal cooling, most experimental data available to date are for rotating rectangular cooling channels with high-performance rib turbulators for Reynolds numbers up to 5×10^4, rotation numbers up to 0.25, and buoyancy parameters up to 0.5. These parameters are applicable for aircraft gas turbines. More studies are needed for the blade-shaped coolant passages (realistic cooling passage geometry, shape, and orientation) with high-performance turbulators and with or without film-cooling holes, for rotating impingement cooling with or without film-coolant extraction, as well as rotating pin-fin cooling with or without trailing-edge ejection to guide efficient rotor blade internal cooling designs. In addition, for land-based power-generation turbines, more studies are needed for rotor coolant passage heat transfer under higher coolant flow (Reynolds number up to 5×10^5), thermal (buoyancy number up to 5), and rotation (rotation number up to 0.5) conditions. Highly accurate and highly detailed local heat transfer coefficient and pressure drop data under these extreme cooling design conditions would be needed to prevent the blade from failure due to local hot spots. Also, study of higher heat transfer enhancement vs lower pressure drop penalty should continue to identify the best heat transfer augmentation technique including compound and new cooling techniques. Development of accurate and efficient CFD prediction tools should continue to provide valuable information for designing effective cooled rotor blades for the new generation of gas turbines.

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